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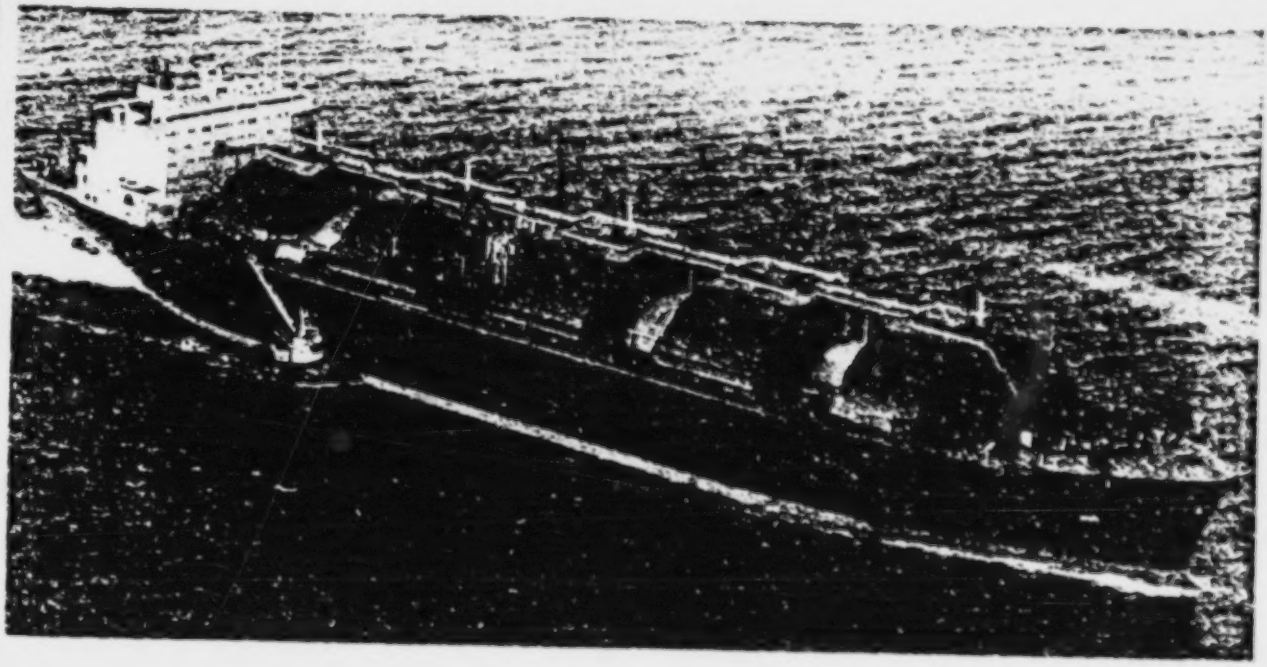

Combined Cycle Propulsion System for LNG Carriers, based on RB-211 Gas Turbine

Prepared for
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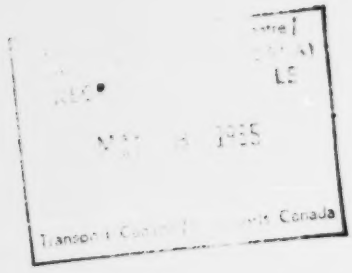
COMBINED CYCLE PROPULSION SYSTEM FOR LNG CARRIERS, BASED ON RB.211 GAS TURBINE

by



A.J. HIGSON

J.A. RIDGWAY

J. ZACHARY

B.B. TEDFORD

ROLLS ROYCE (CANADA) LIMITED,

MONTREAL

CANADA

JULY, 1984.

The contents of this report reflect the views of the contractor and not necessarily the official views or policies of the Transportation Development Centre.

PROJECT TEAM

The project team for Rolls-Royce Limited was composed of:

Mr. I.T. Bowen

Mr. L.G. Brown

Mr. M.J. Neeves

Mr. J.A. Ridgway

Mr. B.B. Tedford

Mr. D.K. Tyler

Mr. J. Zachary

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	C. Alfred Versailles

17. Abstract

This report discusses the performance, costs and Canadian content of a proposed combined gas and steam cycle (COGAS) propulsion system for LNG carriers, based on the Rolls-Royce Marine RB.211 gas turbine. These were then compared with a marine reheat steam propulsion plant for the same duty.

Based on the requirements of a proposal to supply Canadian LNG to Japan (Western LNG Project), performance parameters of a marine propulsion system were generated using a proven gas turbine waste heat software package. From these, equipment specifications for the steam portion of the system were drawn up and distributed to a number of interested manufacturers for estimates of cost and Canadian content. Using their responses, a number of COGAS and "all steam" machinery arrangements were chosen and analysed.

All COGAS plants except the "small boiler" design were found to be over 40% more efficient than the reheat steam plants. Acquisition costs for both types of plant were found to be very nearly equal. Fuel cost savings of between one and two million dollars per year were estimated, depending on type of "make-up" fuel used. Manning requirements are nearly 50% less for COGAS plant. Canadian content can run as high as 50% for the gas turbine, 85% for the waste heat recovery unit and 100% for the steam turbine.

Combined cycle power for LNG carriers was found to be very competitive with reheat steam plant propulsion in first cost and superior to it in fuel cost and Canadian content.

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	C. Alfred Versailles

17. Résumé

Ce rapport présente un système propulsif à cycle combiné pour méthaniers la turbine à gaz utilisée étant une Rolls-Royce RB.211 (version marine). Il compare ce système sous le rapport des performances attendues, du coût et du contenu canadien, à un système de propulsion navale classique à vapeur à resurchauffe.

Partant de la fiche technique du méthanier proposé pour le transport du GNL vers le Japon (projet Western LNG), les performances attendues du système propulsif nécessaire ont été définies grâce à un modèle informatique éprouvé, capable de figurer le cycle de fonctionnement d'un ensemble turbine à gaz/chaudière de récupération.

L'analyse a révélé que tous les systèmes à cycle combiné, sauf celui dit "à chaudière de faible puissance", affichent un rendement de 40 p.cent supérieur à celui des systèmes à resurchauffe, que les deux types de systèmes nécessitent des dépenses de premier établissement pratiquement égales et que les premiers dégagent une économie d'exploitation (combustible) de 1 à 2 millions de dollars selon le type de combustible d'appoint utilisé. Quant au personnel de conduie de la machinerie de propulsion, le système à cycle combiné autorise une réduction de presque 50 p.cent. Enfin le contenu canadien de ce système pourra atteindre 50, 85 et 100 p.cent pour ce qui concerne la turbine à gaz, la chaudière de récupération et la turbine à vapeur respectivement.

18. Mots-clés	19. Diffusion
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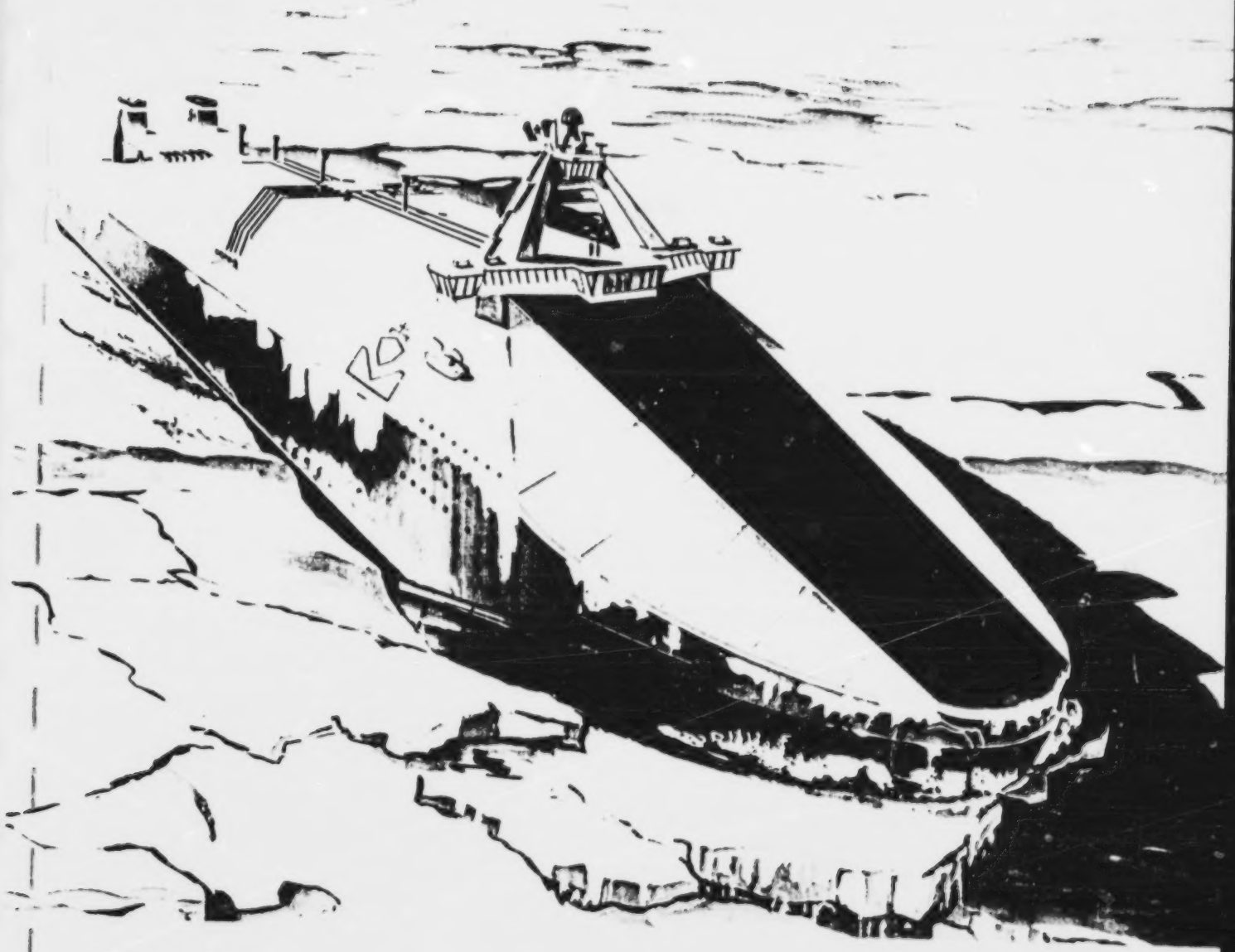


Arctic Resources Carrier
Possible Application for Combined Cycle Propulsion System

1. INTRODUCTION

This is the final report which completes the contract on 'Combined Cycle Study for LNG Carriers' number 6058 commissioned by the Transportation Development Centre of the Canadian Government Department of Transport.

1.1 Propulsion Systems Considered in Study

This study is based on a prime mover gas turbine, incorporating waste heat recovery in the form of a steam boiler, placed within the gas turbine exhaust duct, and associated steam turbine plant geared to the propeller shaft.

An 'all-steam' turbine driven system is included in the study for the purpose of comparison.

The first propulsion system discussed is based upon a combined gas-steam cycle referred to as STAG or COGAS cycle, which, for theoretical evaluation, is based on the Joule or Brayton cycle for the gas turbine and the Rankine cycle for the steam component. The Rankine cycle is used for the 'all-steam' plant comparison.

The principal propulsion unit in the COGAS system is the Rolls-Royce (Canada) Limited Marine RB.211 gas turbine. Both the COGAS and the 'all-steam' plant comparison are based upon a single shaft mechanical drive arrangement. The main reduction gear and transmission components are excluded other than the characteristics of this machinery relevant to performance evaluations.

Diesel engines for marine propulsion are excluded from this study because the useful consumption of cargo boil-off gas by the main engines is an essential requirement. The use of boil-off gas with the present diesel engines results in unsatisfactory fuel consumption relative to their specific power.

However, two small diesel generating sets (each of 2,5 MW capacity), using diesel fuel, are considered for emergency standby for the ship's electrical services for the COGAS and 'all-steam' systems.

1.2 Objectives

1.2.1 Overall

To provide a basis for improved propulsion efficiency and effectiveness of operation of large LNG carriers for the future.

1.2.2 Project Objectives

To make engineering and cost comparisons for the provision of:

a) A Technically feasible and economic propulsion system

design for large LNG carriers of the future employing the RB.211 marine gas turbine in 'Combined Cycle' and to compare this design with the conventional steam propulsion plant.

b) To provide standby propulsion power, by means of ancillary machinery or waste heat boiler refiring, which will conform to the safety requirements of the regulatory authorities.

c) To analyse the alternative plant installations and assess the operational merits.

d) To maximise the Canadian content through design, component procurement, manufacture build and product support.

e) To analyse the relative plant installations and operational advantages.

f) To prepare a budget cost which takes into account acquisition, annual operating and through costing for each plant.

1.3 Programme of Work

The study has been done in five main phases as follows:

Phase 1 Concept Definition

Phase 2 Preliminary Combined Cycle Optimisation

Phase 3 Engineering Enquiry Specification issue and response assessment

Phase 4 Ship performance and operation assessment

Phase 5 Budget cost study

N.B. The interim report, NO. MD.0196, summarised the activity of Phase 1. All phases are discussed in this Final Report.

1.4 Report Completion Time

The original time scheduled for completion of the report was six months beginning September 1983, but more time, extending to June 1984, was needed in order to complete the steam plant manufacturers' submissions.

1.5 Prime Mover (General)

The RB.211-24C/RT56 gas turbine is a marinised version of the aircraft derivative engine first designed for industrial use which has now accumulated over 361 714 hours running time on 41 land based and offshore applications.

The RB.211-522 version of the engine was the lead engine on the Lockhead L1011 Tristar passenger aeroplane and also powers a share of the Boeing 747 fleet in the RB.211-524 form. The industrial version entered service in 1974 at a base load rating of 19,83 MW with an efficiency of 34%. This installation is for gas pumping duty with Trans Canada Pipelines Limited at Burstall, Alberta. The twenty units installed in Canada have accumulated a total exceeding 200 000 operating hours. The remaining units located in Alaska, Mexico, Venezuela, Western Europe and the North Sea increase this figure to 361 714 hours. The latest version of the industrial and marine RB.211 is the RB.211-24C which takes advantage of the latest D4 version of the aero RB.211-524 engine to produce a base load rating of 22,37 MW with and efficiency of 35%. All industrial and marine RB.211's are manufactured and assembled at Rolls-Royce Canada's works in Montreal.

The Marine RB.211-24C/RT56 gas turbine is capable of burning 100% LNG boil-off, 100% distillate oil, or any combination of the two fuels between 90% gas - 10% oil and 10% gas - 90% oil.

1.6 Background

Gas carriers have been traditionally powered by steam turbines where steam raising boilers have made use of the cargo gas boil-off as a fuel. Diesel engines have also been considered for this application but the case is less apparent. The thermal efficiency of the diesel engine, particularly slow speed engines, has increased to 47% (0,178 kg/kW h) using residual or Bunker 'C' fuel oils.

Reliquefaction of LNG boil-off becomes imperative when diesel engines are used for main propulsion. 100% reliquefaction units would be very expensive for present day boil-off rates and are as yet untried on LNG ships. The technical assessment of such a plant is very complex, particularly with regard to plant maintenance life and overhaul.

Recent diesel propulsion studies by HDW, Nippon Kokan and Moss Rosenberg have shown fuel and freight cost advantages over the conventional steam plants but do not consider the full potential of the gas turbine combined cycle plant.

Whether LNG boil-off should be consumed as a fuel or reliquified is a further point of contention. This depends on the relative cost and costing methods of the LNG and the Bunker 'C' or residual fuel oil. Bunker fuel or residual fuel oil is usually the cheaper but against this must be considered the cost of liquefaction plant and the attendant maintenance cost.

Gas burning slow speed diesel engine designs have been attempted, but so far without success. Medium speed gas burning diesels are now available with a dual fuel capability but most of these units require a continuous diesel fuel pilot. Additionally, these engines have reduced thermal efficiencies which would be no greater than 38%.

This report examines the gas turbine combined cycle (COGAS) system, based on the RB.211 gas turbine, and compares its capital and running costs with those of a high efficiency reheat steam plant.

1.7 Efficiency of Marine Propulsion Systems

Marine propulsion plant efficiencies are still improving but more slowly than twenty or so years ago. The benefits that accrue from normal component improvements have ceased to the extent that significant improvements in performance can only be achieved with increases in complexity. Such improvements on steam plants require reheat cycles, regenerative feedheating, and air heater systems.

Gas turbine plants can achieve significant improvements in efficiency with the use of exhaust tail cycles or intercooling and regenerative cycles.

2. CONCEPT DEFINITION (Study Phase 1)

2.1 General

This was the initial phase of the study which started in September 1983 with the intention of providing a basis for meeting the objectives of the study as stated in Section 1.3.

The main parts of this phase were as follows:

- Data gathering for the basic ship design operating profile and supplier options.
- Definition of the maximum gas turbine rating and collection of performance data.
- Preliminary definition of reheat steam plants and data collection from selected suppliers.
- Preliminary plant installation scheme production.
- Matching of plant and ship power requirements for initial setting of the gas turbine rating.

The results which provided the basis and agreement for the study were submitted to TDC in an interim Rolls-Royce report No. MD.0196 dated October, 1983.

The main parts of this report defined the basis for the ship design, operation, gas turbine performance and ship machinery layouts, which then provided the foundation upon which supplier enquiry specifications were drawn up.

2.2 Ship Design and Operation

2.2.1 Application

The design of ship used for this study is based on that featured in the Dome Western LNG Project. A standard LNG cargo capacity of 125 000 m^3 has been used, contained in five spherical tanks at -163 deg.C. The system of containment is similar to that employed in the Moss Rosenberg vessel.

2.2.2 Design

The vessel will be operated at an economical service speed calculated to satisfy the prospective delivery schedule using five ships. An estimated hull resistance curve for such a ship is shown in Fig. 1.

The main characteristics of the hull and propulsion machinery requirements are as follows:

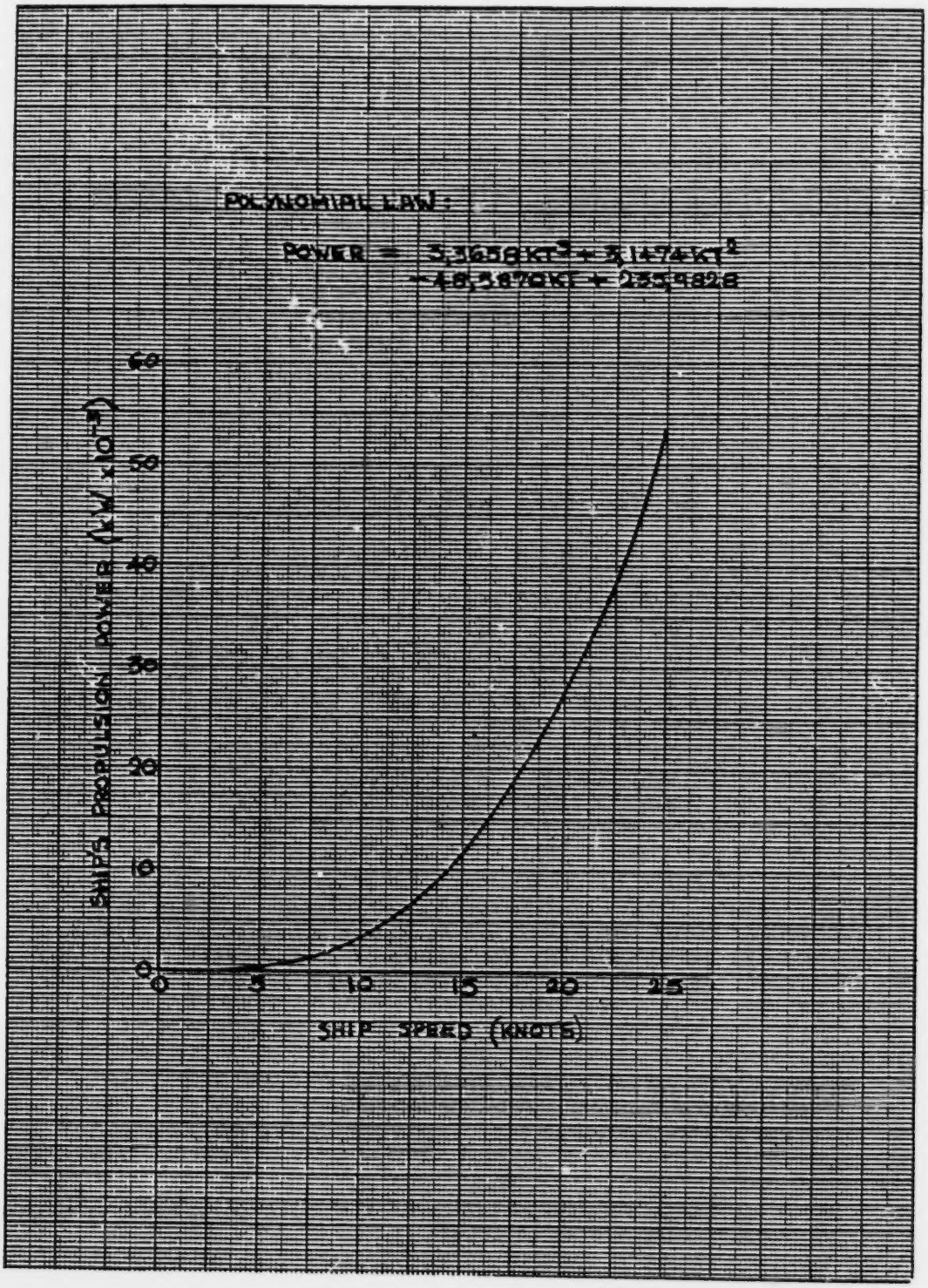


Figure 1 LNG CARRIER HULL RESISTANCE CURVE

Loaded displacement tonnes	100 000 Cargo
Weight tonnes	54 000
Loaded speed of vessel (outward) knots	17,5
Ballast speed of vessel (return) knots	17,5
Loaded propulsion power at shaft kW	18 387
Ballast propulsion power at shaft kW	16 916
Reduction gear efficiency (assumed constant)	97%
Loaded propulsion power at gearbox kW	18 956
Ballast propulsion power at gearbox kW	17 440

Main Machinery Requirements:

a) COGAS

The gas turbine combined cycle plant will comprise one Marine RB.211 propulsion unit together with a waste heat recovery boiler (WHRB) and steam turbine. In addition to providing shaft power for ship propulsion the boiler will also supply ship's services power and steam for LNG maintenance. A variable or controllable pitch propeller (CPP) is fitted to ensure that propulsion efficiency is maintained as well as providing excellent ship manoeuvrability.

Under normal steady state operation total propulsive power will be supplied by the COGAS plant. In the unlikely event of gas turbine shutdown significant propulsion power is available from the steam turbine by refiring the WHRB. If the WHRB is inoperative then steam can be supplied from the dual fuel auxiliary boiler.

b) 'All-Steam' Plant

The steam plant is a conventional design comprising a main boiler, steam turbine and associated condenser and feed pump arrangements. A fixed pitch propeller (FPP) and non reversing gearbox is envisaged with a reverse steam turbine for astern operation and manoeuvring. Shaft driven generators will provide power for ship's services. If the LNG burning main boiler should fail then an auxiliary 'take-home' boiler (dual fuel) will supply sufficient propulsion steam to the main turbine in addition to meeting usual 'at sea' steam requirements for ship's services.

2.2.3 Ship's Services Requirements

The ship's power demand is met by the steam in two ways. In the case of the COGAS plant, steam produced by the superheater will supply the steam turbine as well as providing steam for LNG vapour fuel heating, inner hull and ship's hotel heating, and desalination via a spray desuperheater. In the case of the 'all-steam' plant a steam bleed will supply heat to the LNG vapour fuel heating and hotel loads.

Shaft driven generators are used in both plants but in slightly different ways. The COGAS plant has a CPP and so will maintain propeller revolutions down to quite slow ship speeds and therefore exhibit a fairly constant generated frequency. When the propeller

speed falls too far an auxiliary turbine generator will take over the electrical duty. The 'all-steam' plant uses a FPP and so will require frequency compensation via a suitable electronic conversion package. Eventually shaft speed will again fall below a level for reliable frequency conversion and auxiliary turbine generators will be brought on-line.

The requirements for ship services are as follows:

Steam production for LNG vapourisation and vapour fuel heating, hotel services and some inner hull and cofferdam heating	4 T/h. @ 340 deg.C. and 1850 kPa

Electrical power:	
At sea	2000 kWe (1x2500 kWe T/G)
Entering harbour (bow thruster)	5000 kWe (2x2500 kWe T/G)
In harbour	
Unloading	3600 kWe (2x2500 kWe T/G)
Loading	2500 kWe (1x2500 kWe T/G)

In harbour, operation will be supported by two 1800 kWe diesel generators where required.

Boil-off compression*	400 kW (2 T/C, 1 standby)

* This is required for the gas turbine plant only - a gas fuel supply of 3450 kPa is required at the fuel system interface.

2.2.4 The LNG Fuel

The boil-off rate for LNG is given as a percentage of the stored volume per unit time at a particular temperature. The variation of boil-off rate with temperature for both the loaded and ballast conditions for a carrier of this type were estimated as shown in Fig. 2.

The LNG component breakdown by volume can be taken as follows:

	%
Methane	97,919
Ethane	1,063
Propane	0,038
Nitrogen	0,980

The gas will provide a fuel lower calorific value (LCV) of 49 138 kJ/kg.

2.2.5 Design and Safety Codes

The hull and equipment are designed and built to the following standards and rules.

- International Convention for the Safety of Life at Sea 1974 and its protocol of 1978.

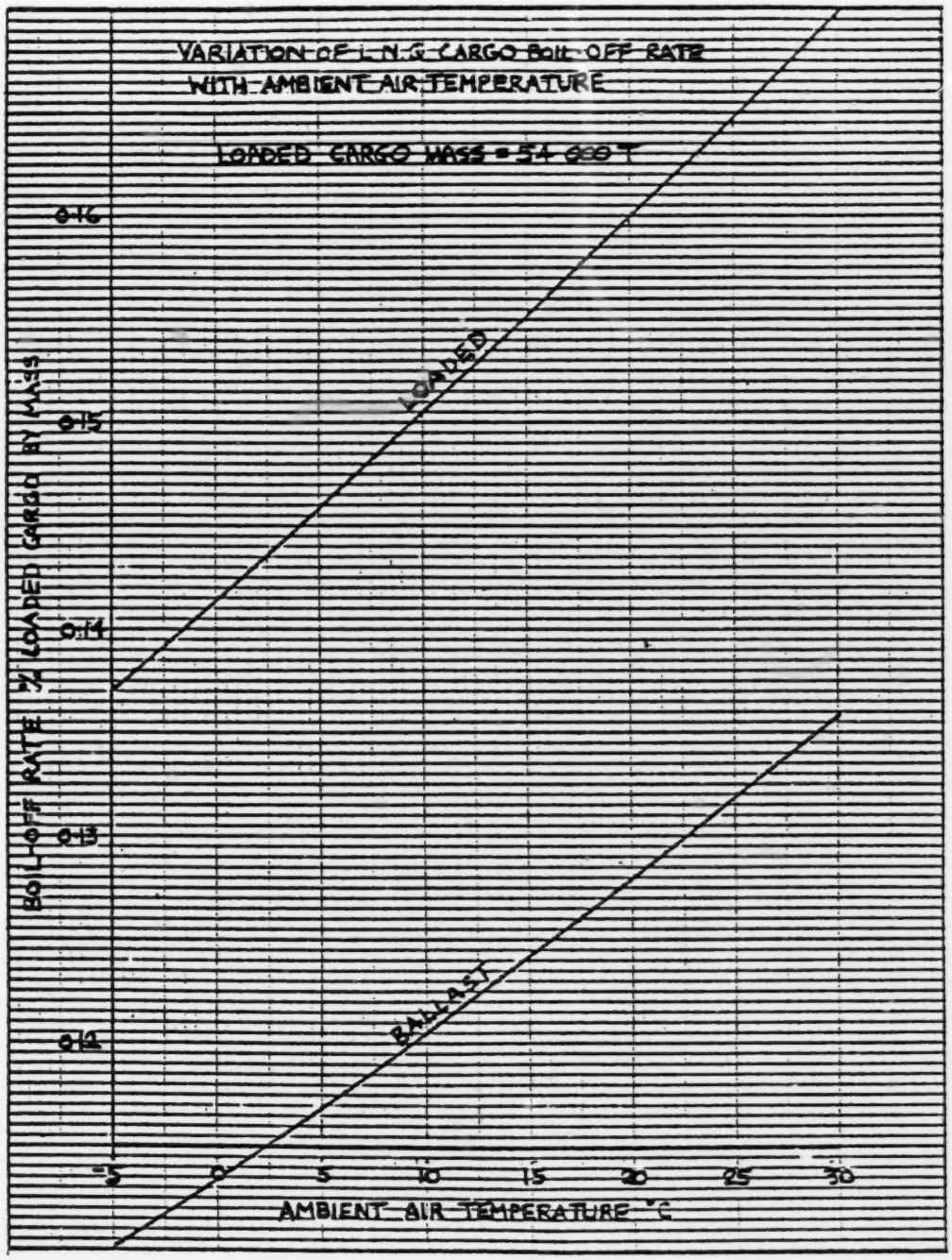


Figure 2 VARIATION OF LNG CARGO BOIL-OFF RATE WITH AMBIENT AIR TEMPERATURE

- International Convention on Load Lines 1966

- International Telecommunication Convention Malage-Torremolinos, 1973

- International Convention for the Prevention of Pollution from Ships 1973, and its protocol of 1978

- IMCO Code for the Construction and Equipment of Ships Carrying Liquified Gases in Bulk (A.328 (IX))

- Canadian Shipping Act

- Canadian National Transportaion Act

- Lloyds 100A1 Liquified Gas Carrier

- Lloyds Machinery Certificate (LMC)

- UMS Certificate

In addition, classifications of the United States and Japanese Ministry of Transport are likely to be sought.

2.2.6 Operation of Vessel

An economical service speed of 17,5 knots was chosen to satisfy the prospective delivery using five ships. These ships are assumed to operate between Prince Rupert, British Columbia and Osaka, Japan. A minimum of 28 days downtime per annum has been allowed for all maintenance, inspections, and surveys for each ship.

Careful consideration of the range of air and sea temperatures for the above route through all seasons of the year yielded an all up average air and sea water temperature of 9 deg.C. The source for this information was the U.S. Navy Marine Climate Atlas of the World - Volume II, which covers the North Pacific Ocean.

On this basis it was decided to fix the ambient air temperature around which the plant would be designed at 9 deg.C. The seawater temperature was also taken to be 9 deg.C. so that an estimate could be made of the sizes of various coolers and condensing equipment.

The highest mean sea surface air temperature likely to be encountered is approximately 26 deg.C. - off the coast of Japan in July. The lowest mean sea surface air temperature likely to be experienced will be approximately 4 deg.C. - off Prince Rupert, British Columbia in January.

Project Delivery Schedule:

Year	Annual LNG Delivered ($kJ \times 10^{12}$)
1987	5,524
1988	14,032
1989	124,280
1990	146,374
1991	151,897
1992-2006	160,183
2007	121,518

Voyage Profile:

Each round trip voyage is assumed to comprise of the following:

a) Low speed approach to Prince Rupert, B.C.	8 hours
b) Waiting if necessary	1 day
c) Manoeuvring	2 hours
d) Loading	1 day
e) Manoeuvring	2 hours
f) West bound loaded leg to Japan	10 days
g) Low speed approach to Japanese terminal	10 hours
h) Waiting if necessary	22 hours
i) Manoeuvring	2 hours
j) Unloading	1 day
k) Manoeuvring	2 hours
l) East bound leg to Canada in ballast	10 days

The above data is shown in Figure 3.

Annual Fleet Operation:

On the basis of the foregoing voyage profile and the cargo boil-off rates at the design ambient temperature, the annual delivery during the period of maximum delivery 1992-2006 will be satisfied by the following:

Loaded cargo boil-off interval - 11 days 18 hours

Ballast cargo boil-off interval - 11 days 6 hours

Ballast volume remaining for return leg voyage is equal to the ballast boil-off rate x length of voyage + 3 days.

Loaded cargo - 54 000 T

Fuel delivered per round trip:

= 54 000 (1-0,0015x11,75-0,0012(11,25+3)) Tonnes

= 52 126 Tonnes

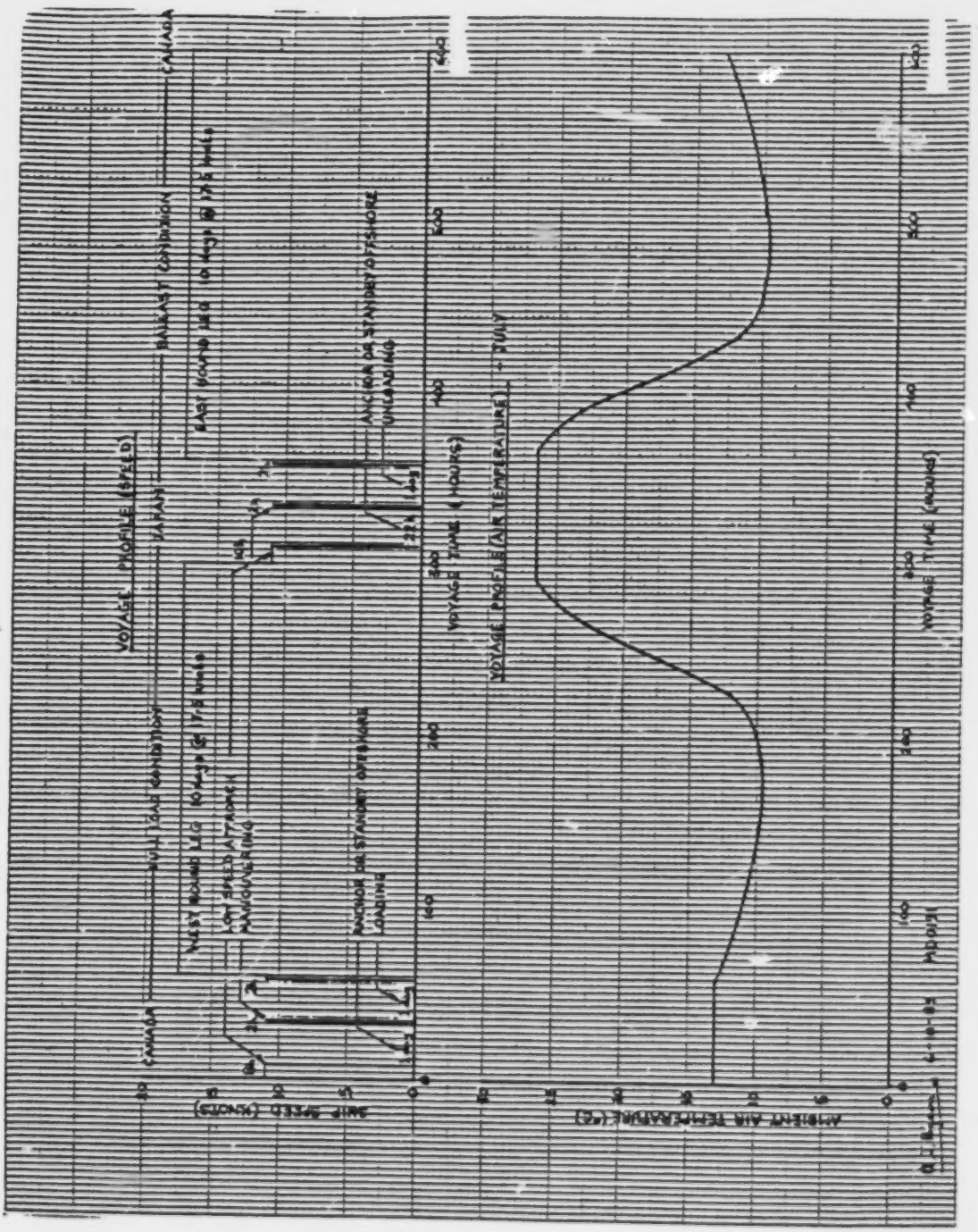


Figure 3 VOYAGE PROFILE

= 52,126x10^6 kg x 49 138 kJ/kg

= 2,5614x10^{12} kJ

Annual delivery requirement in 1992-2006 = 160,18x10^{12} kJ

Total annual round trips required = 62,54

Average trips/ship/annum (5 ships) = 12,5

Average annual service time per ship = 313 days

For simplicity a figure of 13 trips/year for the ship will be used for estimation of plant and component life and reliability.

2.3 Gas Turbine Performance and Preliminary COGAS Design

2.3.1 Prime mover - brief description

The marine RB.211 propulsion unit comprises the high efficiency twin spool gas generator and long life two stage RT.56 free power turbine, both units being encased within an acoustic enclosure and mounted upon a rigid steel baseframe

- Gas Generator

The marine gas generator is a derivative of its aero counterpart with the fan and the LP turbine removed. The unit has a seven stage intermediate pressure (IP) stage compressor, a six stage high pressure (HP) compressor with a single stage HP and a single stage IP turbine. The designation IP has been retained for commonality with aero terminology inspite of the removal of the aero LP section.

A single stage of variable incidence inlet guide vanes, in conjunction with bleed valves on both compressors maintains optimum performance over the whole speed range with minimum mechanical complexity

Bleed valves on the compressor assist low power starting and give good handling characteristics.

Bleed points on the casings provide IP and HP compressor air for power turbine cooling and other services.

Combustion

There is an annular combustion chamber with 18 burners. The burners are designed to function with gaseous fuel, distillate liquid fuel, or a mixture of the two. The annular design of the chamber provides a clean aerodynamic extension of the HP compressor outlet, minimises the area to be cooled, gives good temperature distribution and avoids interconnector problems.

Bearings

Bearings are positioned away from the hot combustion zone. Marine roller bearings are hydraulically clamped by a squeeze film which is a thin film of oil under pressure between the bearing outer track and bearing housing. This squeeze film minimises vibration, increases component life and ensures correct operation of the bearings' rolling elements at all loads and shaft speeds.

The RB.211 gas generator is constructed from five modules which are fully interchangeable with standard replacements. Modules may be changed quickly on board ship and hence down time and maintenance costs are minimised.

- The RT56 Power Turbine

The RT56 is a two stage power turbine with a moderate energy conversion per stage and very high blade efficiencies. Power turbine disc cooling is provided by an air bleed from the gas generator IP compressor. The air is routed via the first stage nozzles on the rim of the disc. Interlock tip shrouds are incorporated to prevent vibration, and with the honeycomb lined casings designed to maintain blade tip sealing, the mechanical integrity and high efficiency of the power turbine are assured.

Tilting pad thrust and journal bearings maintain maximum rotor dynamic stability.

2.3.2 RB.211 Performance

A power turbine speed of 4800 rev/min was selected to allow the best economy of operation at high and low powers while at the same time providing an overspeed margin sufficient to accommodate the transmission speed fluctuations likely to be experienced in heavy seas. The following performance data, shown graphically in Figures 4 to 7, relates to installed engine performance with a level of exhaust back pressure in excess of that normally experienced in a typical ship installation with uninterrupted exhaust flow.

There are two performance ratings available for use with base load merchant ships such as LNG carriers.

Maximum Continuous Rating - the highest power which is recommended for use in merchant ship applications and is available for short durations and emergency use (up to 10% of the total running time).

Service rating - the recommended power at which the engine can be operated continuously in a merchant ship application in order that long life between inspection periods may be obtained.

It should be appreciated that the above ratings represent envelopes within which the engine may be operated and are themselves conservative in comparison with those for industrial applications in

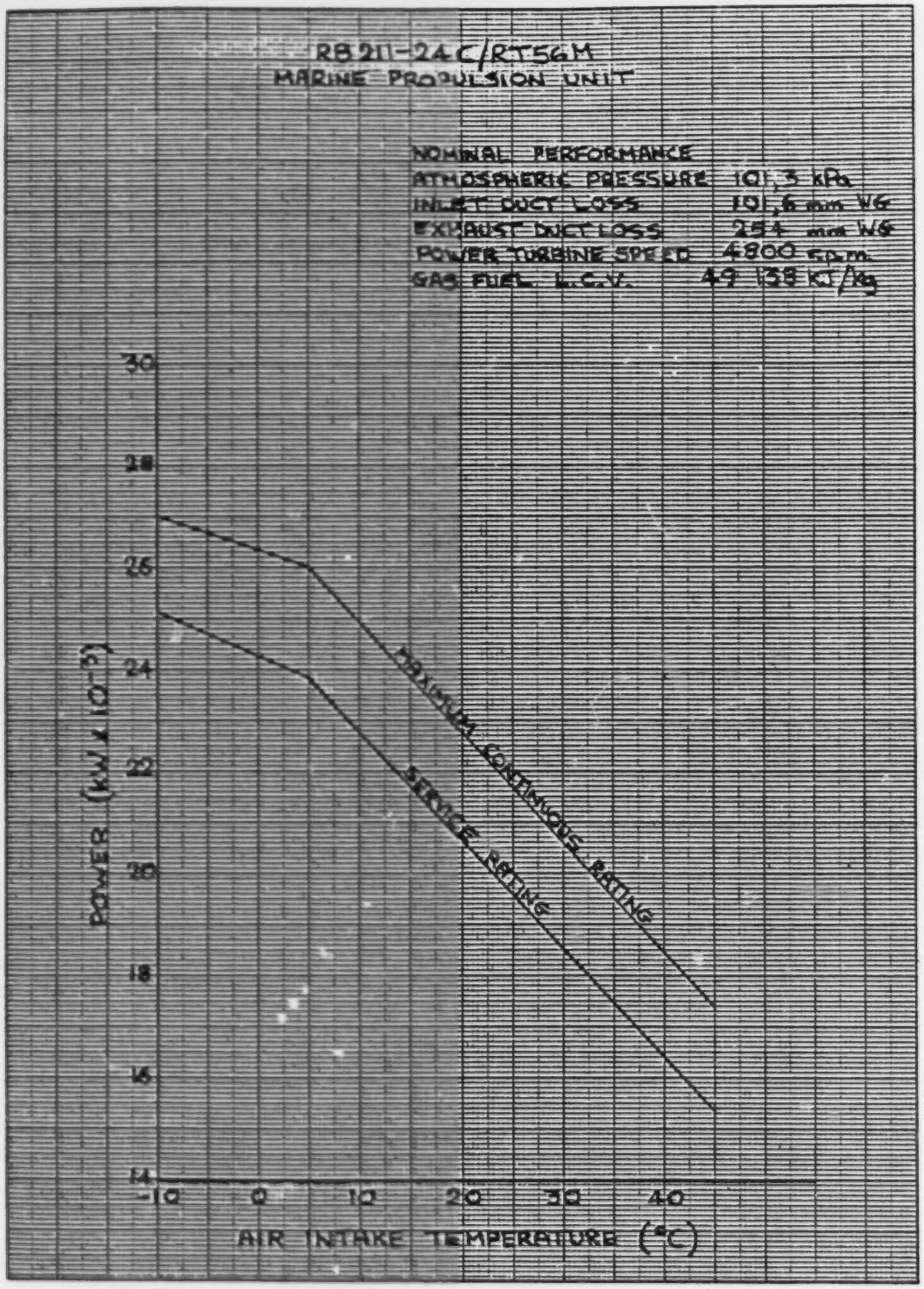


Figure 4 VARIATION OF POWER WITH AIR INTAKE TEMPERATURE

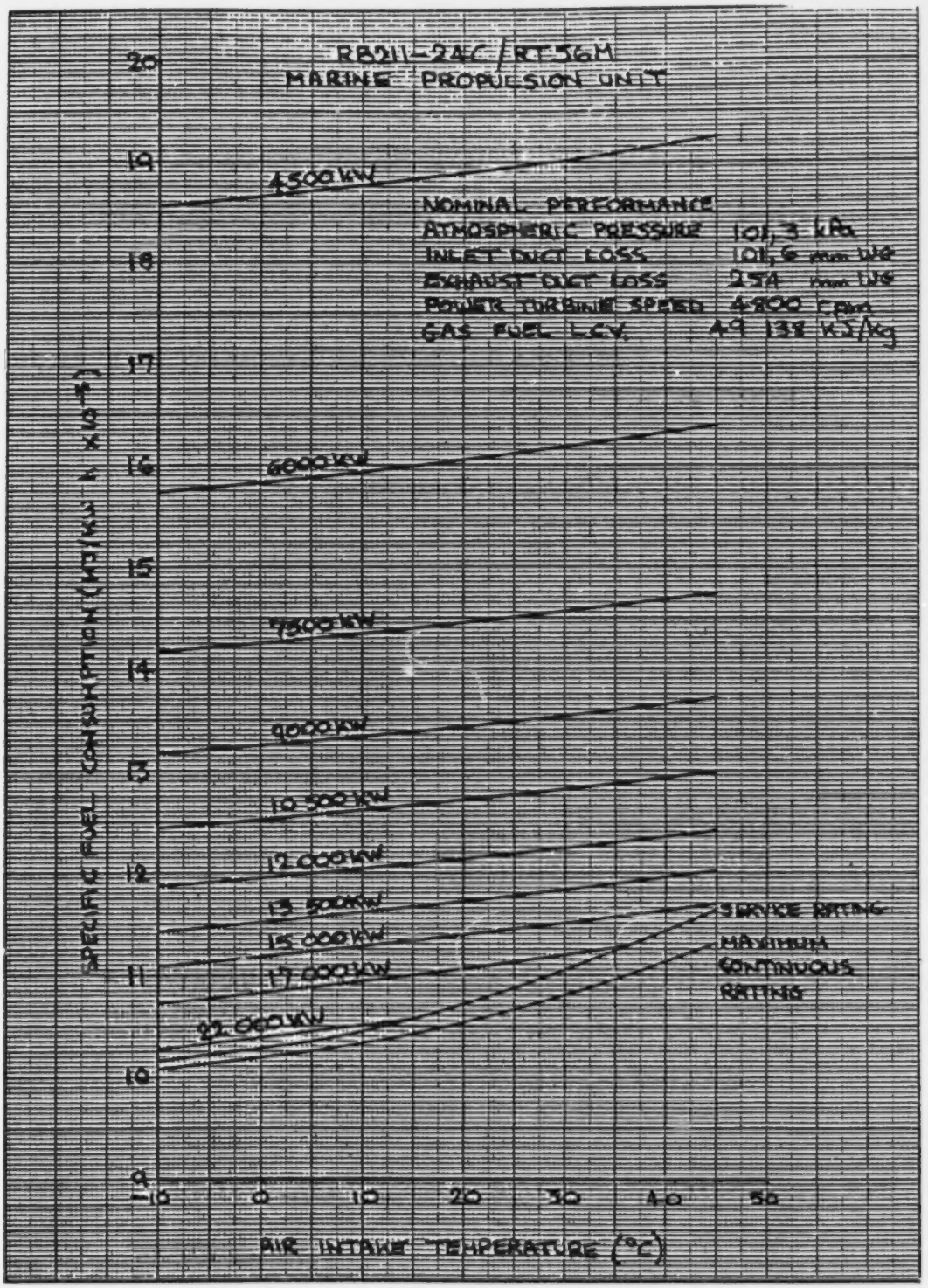


Figure 5 VARIATION OF SPECIFIC FUEL CONSUMPTION WITH AIR INTAKE TEMPERATURE AT FIXED POWERS

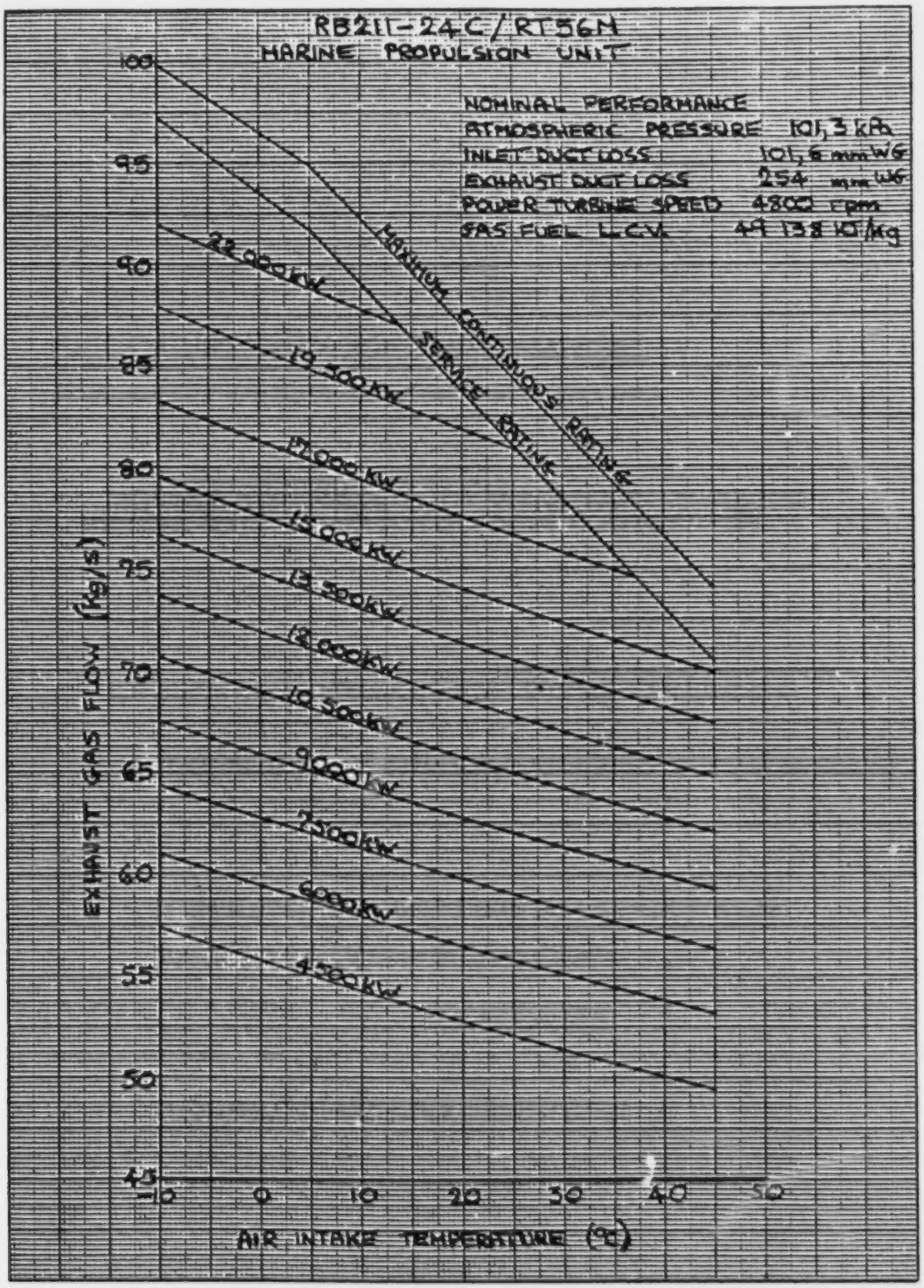


Figure 6 VARIATION OF EXHAUST GAS MASS FLOW WITH AIR INTAKE TEMPERATURE AT FIXED POWER

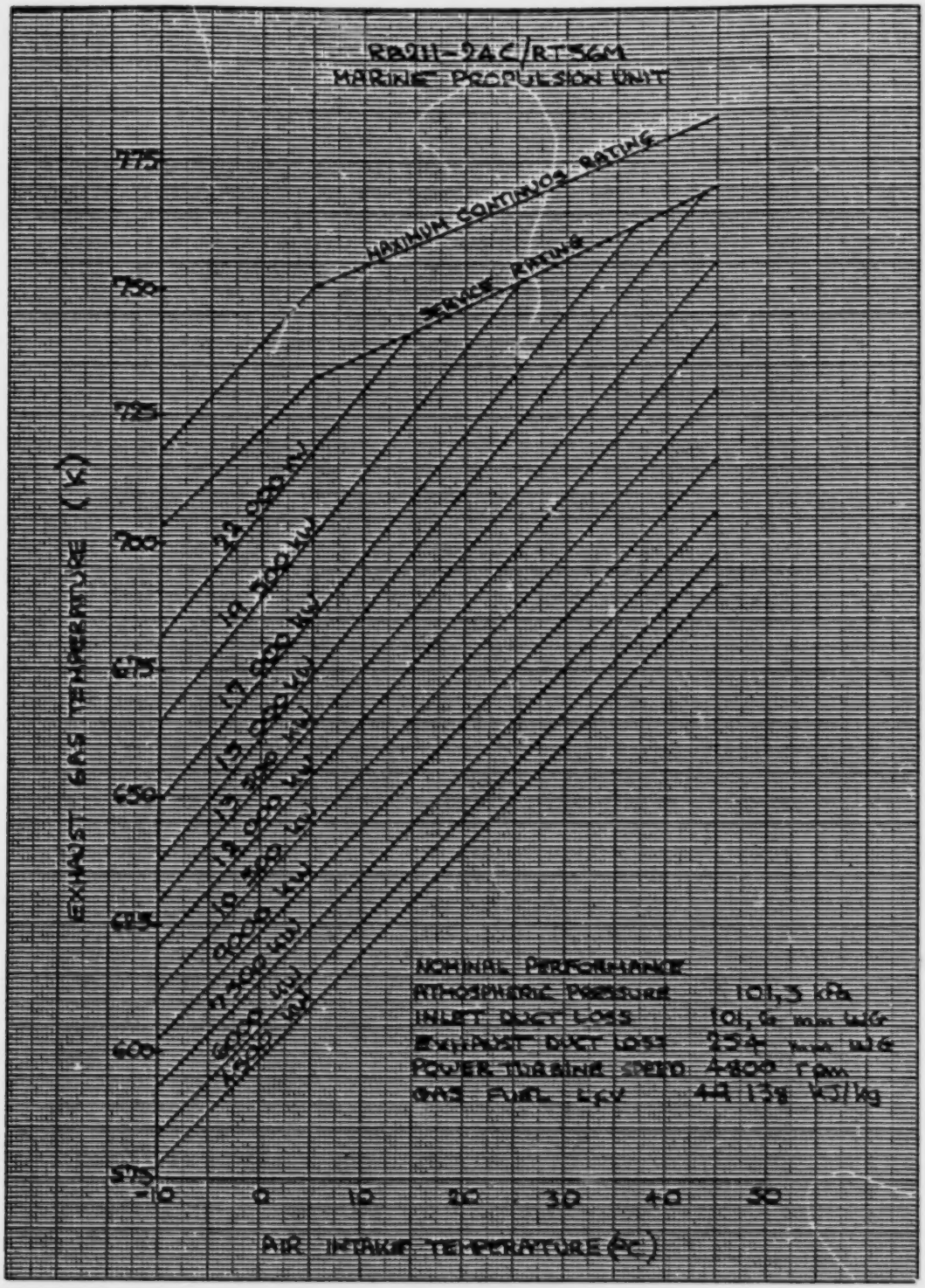


Figure 7 VARIATION OF EXHAUST GAS TEMPERATURE WITH AIR INTAKE TEMPERATURE AT FIXED POWERS

order to ensure the high degree of reliability which Rolls-Royce Limited considers essential for such applications. The performance is based on nominal gas generator and power turbine data. If differences arise between the performance contained in this document and that published in any subsequent contractual document then the latter document will be overiding.

The basic engine assumptions are as follows:

Atmospheric pressure of 101,3 kPa

Engine anti-icing switched off

No air bleed from the engine other than that released by the compressor blow-off valves

Intake duct pressure loss of 101,6 mm W.G.

Exhaust duct pressure loss of 254 mm W.G. (includes waste heat boiler estimated blockage effect)

Gas fuel LCV of 49 138 kJ/kg

2.3.3 Prediction of Initial Gas Turbine Power Setting

Taking the power required at the propeller shaft and combining this with the expected ship services demand for the 'at sea' design condition of 17,5 knots, a total combined cycle mechanical power of 21 404 kW was identified.

$$\frac{\text{Shaft Power}}{\text{Gearing Efficiency}} + \frac{\text{Electrical Power}}{\text{Electrical Efficiency}}$$

$$+ \frac{\text{Gas Compression Power}}{\text{Electrical Efficiency}} = \text{Total Mechanical Power}$$

$$\frac{18\ 387}{0,97}\ \text{kW} + \frac{2000}{0,98}\ \text{KWe} + \frac{400}{0,98}\ \text{kW} = 21\ 404\ \text{kW}$$

For a typical waste heat recovery boiler and turbine plant overall efficiency increases with decreasing working pressure. As the working pressure is reduced for a given output power the boiler area will increase. Any recovery plant will then be a result of consideration of the practical balancing of efficiency, against size of plant and hence cost.

Using a proven gas turbine waste heat recovery software package developed by Rolls-Royce it was possible very rapidly to consider the effects of varying boiler pressure.

A boiler pressure of 2600 kPa was finally settled upon as a suitable operating pressure for an application of this type where the waste heat boiler and steam turbine are required to be compact units. By running the simulation program again, but now taking into

account the constant steam services bleed, it was possible, by varying the gas turbine setting, to isolate a gas turbine power which satisfied the total power requirement. This assumed that electrical services and compression duty would be taken off via a shaft driven generator system.

The resulting initial gas turbine throttle setting was 16 816 kW at the design air temperature of 9 deg.C.

- Supplementary Firing

When supplementary firing is used in combined cycle plants additional fuel is burned in the gas turbine efflux to raise its temperature and so increase its energy content before it passes through the WHRB. This increases the effectiveness of the boiler in recovering heat, improves the steam conditions and generates more steam.

By comparing equations of overall thermal efficiency for supplementary fired and non assisted waste heat recovery boilers it can be shown that the former will in fact have a lower overall efficiency. The boiler efficiency is improved as expected but at the expense of a drop in the overall efficiency of the system. This result is in accordance with the thermodynamic principle that, for maximum efficiency, heat should be added at the highest temperature in the cycle.

More power can be produced per unit size by supplementary firing but at the same time increased gas temperatures at entry to the boiler may mean a more elaborate and expensive material specification.

- Effect of Gas Turbine Efficiency on Overall Cycle Efficiency

It can be shown that for a gas turbine combined cycle plant without supplementary firing the following equation is true.

$$\eta_{CC} = \eta_{GT} + (1-\eta_{GT}) \cdot \eta_{TC} \qquad \text{Equation 1}$$

where:

η_{CC} = combined cycle plant thermal efficiency

η_{GT} = gas turbine simple cycle thermal efficiency

η_{TC} = steam cycle (tail cycle) overall thermal efficiency in the gas turbine exhaust

$(1-\eta_{GT})$ = the total energy rejected (ignoring casing radiation and lube oil losses which are assumed to be negligible in comparison)

From equation (1) it can readily be seen that a lower gas turbine simple cycle efficiency will result in more heat being

rejected in the exhaust gases per unit fuel consumption. This means that a combined cycle plant employing a gas turbine with a poor thermal efficiency will need a much larger and more efficient recovery plant than would a plant using a high efficiency gas turbine for the same combined cycle efficiency.

A high efficiency combined cycle plant based on a poor gas turbine efficiency will mean a complete system not much different from the equivalent 'all-steam' option with the added complication of two types of machinery.

- Tail Cycle Component Efficiencies

Let η_B = boiler efficiency
$\eta_{T\&C}$ = steam turbine and condenser efficiency

To minimise the chance of dew point related corrosion problems in waste heat boilers it is sensible to work on an expected recovery from gas to steam (η_B) of about 60%, 70% at the most.

The 'net' cycle effiency of a simple modern steam turbine and condenser will be about 27% ($\eta_{T\&C}$). Therefore the effective tail cycle efficiency will be:

Equation 2

$$\eta_{TC} = \eta_B \times \eta_{T\&C} = 60\% \times 27\% = 16{,}2\%$$

when there are no service steam bleeds

Figure 8 shows the variation of both η_B and $\eta_{T\&C}$ efficiencies with changing gas turbine power setting for a continuous ship speed of 17,5 knots. Account was taken of steam bleeds, ship services load and electrical boil-off compression duty (η_B falls in respect of turbine output when steam bleeds are used)

Figure 9 shows the variation of combined cycle efficiency with gas turbine efficiency for a given tail cycle efficiency of between 10% and 18%.

For example, this graph shows that for a gas turbine with a simple cycle efficiency of 0,3315 operating in conjunction with a steam plant of tail cycle efficiency = 11% will produce a combined cycle efficiency of 0,3433. If a gas turbine of 0,3218 simple cycle efficiency is used then a more efficient steam plant with a tail cycle efficiency of 13% would be required to maintain the level of combined cycle efficiency at 0,3433. These results are drawn from a more complex form of equation (1) which includes allowances for steam bleeds, services and boil-off compression (equation (1) Fig. 9). With services' steam bleeds, as much as 13% of the exhaust energy can be turned into steam turbine power.

Design against Corrosion

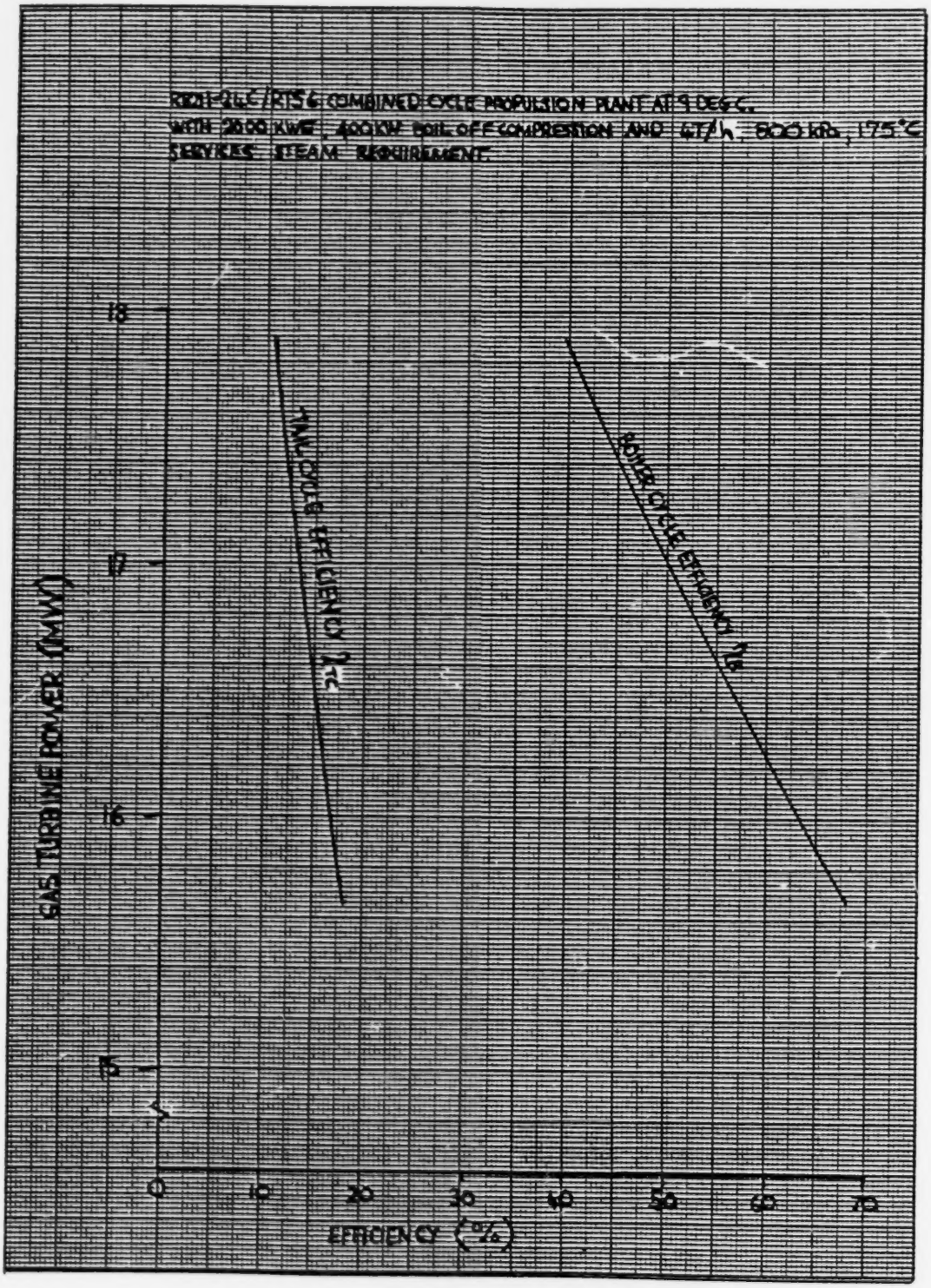


Figure 8 VARIATION OF TAIL CYCLE AND BOILER CYCLE EFFICIENCY WITH GAS TURBINE POWER FOR A CONSTANT SHIP SPEED OF 17,5 KNOTS AT 9 DEG.C. AIT.

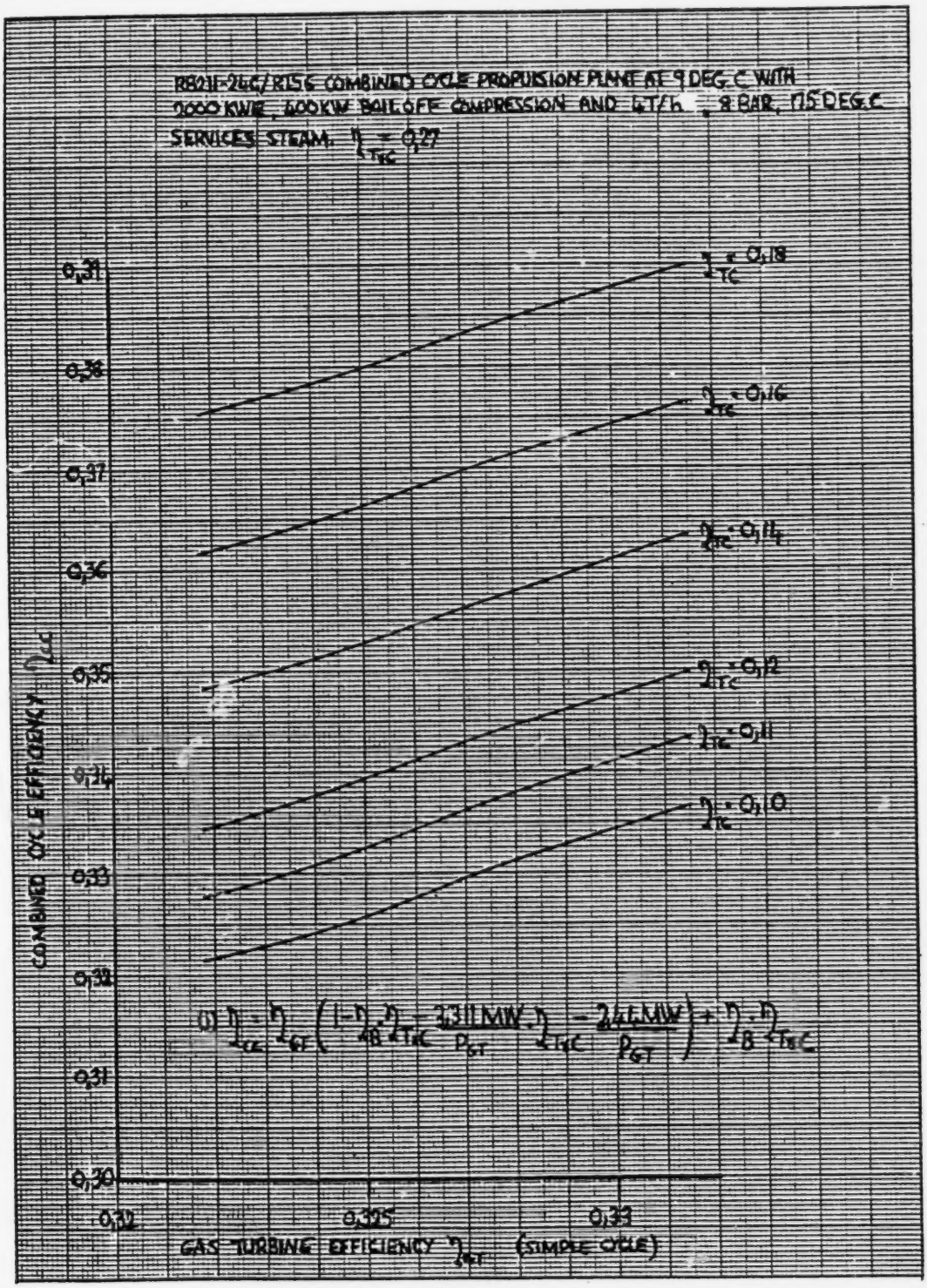


Figure 9 VARIATION OF COMBINED CYCLE EFFICIENCY WITH GAS TURBINE EFFICIENCY FOR A GIVEN TAIL CYCLE EFFICIENCY (CONSTANT S.TURBINE AND CONDENSER EFFICIENCY)

The boiler will be subject to both internal and external corrosion. Internal corrosion can be prevented by treating the feed water to remove or neutralise the various impurities, particularly dissolved oxygen and carbon dioxide gases. This can be done by thermo-mechanical and/or chemical means although excessive addition of solids during feedwater treatment is not recommended.

External corrosion in the boiler can occur on the economiser tubes if the surface temperature of the pipework is at or falls below the dewpoint of the gas turbine exhaust when the engine is running on distillate fuel. Sulphuric acid corrosion can thus be eliminated by ensuring a sufficiently high economiser return temperature relative to the exhaust gas temperature in the boiler. Corrosion will not be a problem when the ship is running on LNG.

Choice of Single or Dual Pressure Boiler

In the interests of simplicity and cost it was decided that the WHRB should be a single pressure unit operating with conventional drum level control and slaved to the conditions of the gas turbine exhaust.

Computer Simulation

The largest sensible boiler was used for the simulation employing a 20 deg.C. pinch and 40 deg.C. approach design. A high pressure steam bleed for ship's services (4T/h) avoided complex heat exchange equipment consideration and was relatively easy to model on existing Rolls-Royce steam cycle software.

The resulting preliminary gas turbine throttle setting was found to be 19 052 kW at the design point temperature giving a fuel rate for the total plant of 0,1564 kg(LNG)/kW h at the turbine couplings. This figure allows for steam services, compression and electrical demands at the design ship speed of 17,5 knots.

2.3.4 Basic Requirements of the COGAS Plant Design
(See Fig. 10 for schematic)

The design will be required to demonstrate flexibility and reliability of power in operation. This will be achieved in the following ways.

i) The WHRB will be capable of operating independently of the gas turbine using a dedicated burner system and damper by-pass arrangement. In the event of a gas turbine shut down LNG will be routed to the WHRB burner so continuing to burn boil-off gas and providing 'take-home' power at the COGAS steam turbine. The steam turbine power developed in the 'take-home' mode will be about 50% more than that during normal operation in order to maintain a satisfactory ship speed. The conditions of the gas path inside the WHRB at 'take-home' using LNG will be equivalent to those existing when the gas turbine is running at its Maximum Continuous Rating. This raises the boiler stop valve pressure from the normal level of

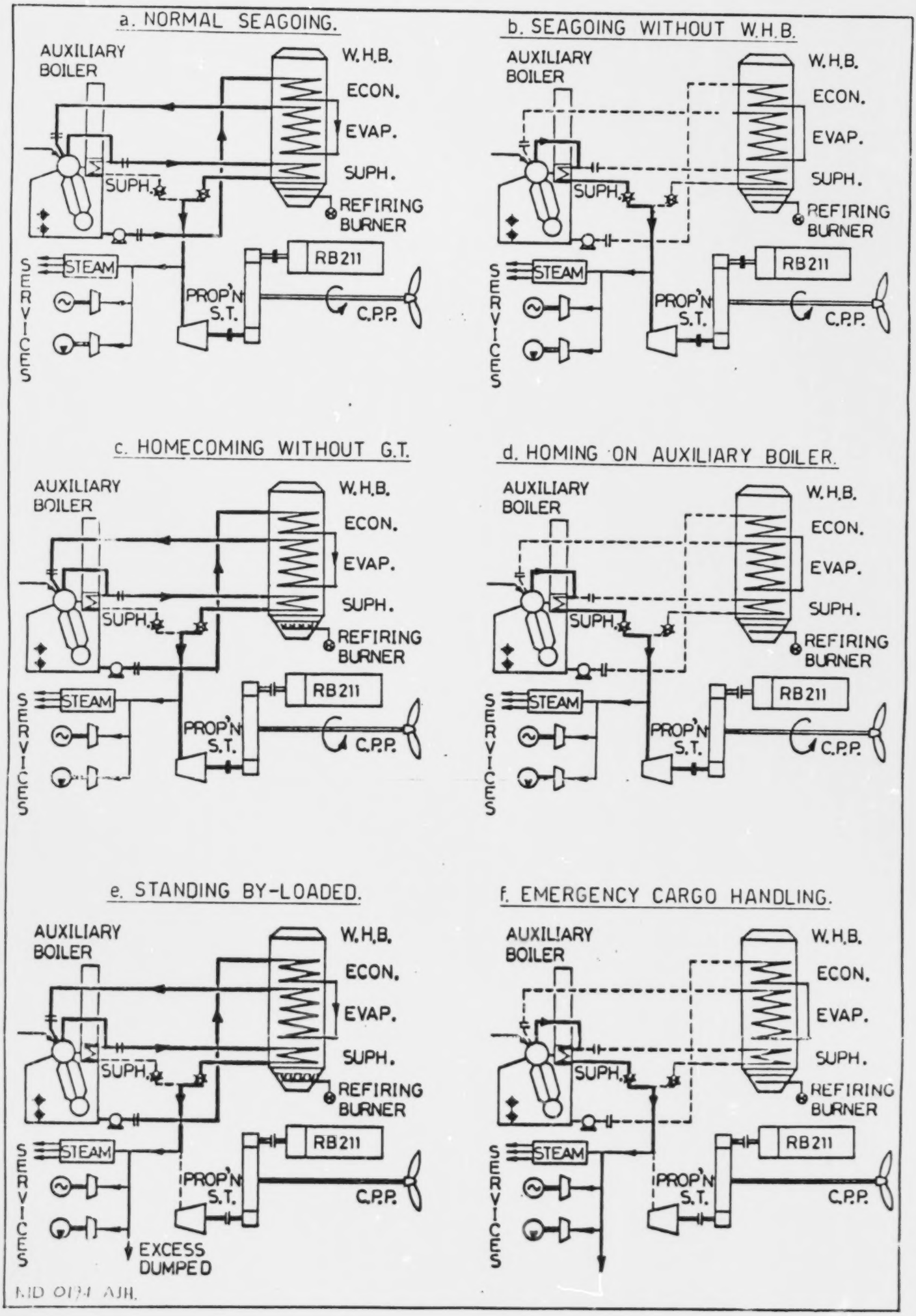


Figure 10 SCHEMATIC - COGAS PLANT

MD 0134 AJH.

2600 kPa to 3900 kPa and the steam turbine power from 4579 kW to 7114 kW.

ii) In the event of both the gas turbine and the WHRB being inoperative a dual fuel auxiliary boiler will provide 'take-home' steam from either LNG or diesel fuel (evaporation approximately 30T/h) as well as continuing to provide ship's services' power. This boiler will be capable of consuming boil-off while in port.

iii) In 'normal' operation the WHRB and steam turbine will operate on a sliding pressure basis without use of turbine admission control. The boiler pressure will follow the rate of steam flow retaining a higher thermal efficiency. In the event of rapid changes in gas turbine throttle setting and hence exhaust gas conditions, such as would be experienced during manoeuvring, throttle control may be re-instated to maintain constant pressure and prevent control lag problems.

Summarising, the envisaged operation options are as follows:

Normal Modes

a) Normal sea going (combined cycle).

b) Stand-by in port or offshore with continuous boil-off consumption (through WHRB and auxiliary boiler).

c) In port or emergency cargo handling using auxiliary boiler and diesel generators or WHRB plant.

Emergency Modes

a) Sea going with WHRB plant shut down (gas turbine only)

b) Home coming with the gas turbine shut down and the WHRB being refired using LNG through its own burners.

The preliminary installation arrangement for the COGAS propulsion machinery in the ships engine space is shown in drawing Figure 11. The five main items of the propulsion system shown in outline, comprise a marine RB.211 gas turbine propulsion unit, single pressure WHRB, condensing propulsion steam turbine, main reduction gear and auxiliary boiler plant. This was an initial estimate of the types of plant we would be considering and was prepared to accompany the Enquiry Document as a submission to participating manufacturers.

2.3.5 Description of COGAS Installation Drawing (Fig 11)

The installation consists of the following:

- RB.211/RT56 module
- Air Intake System
- Exhaust System

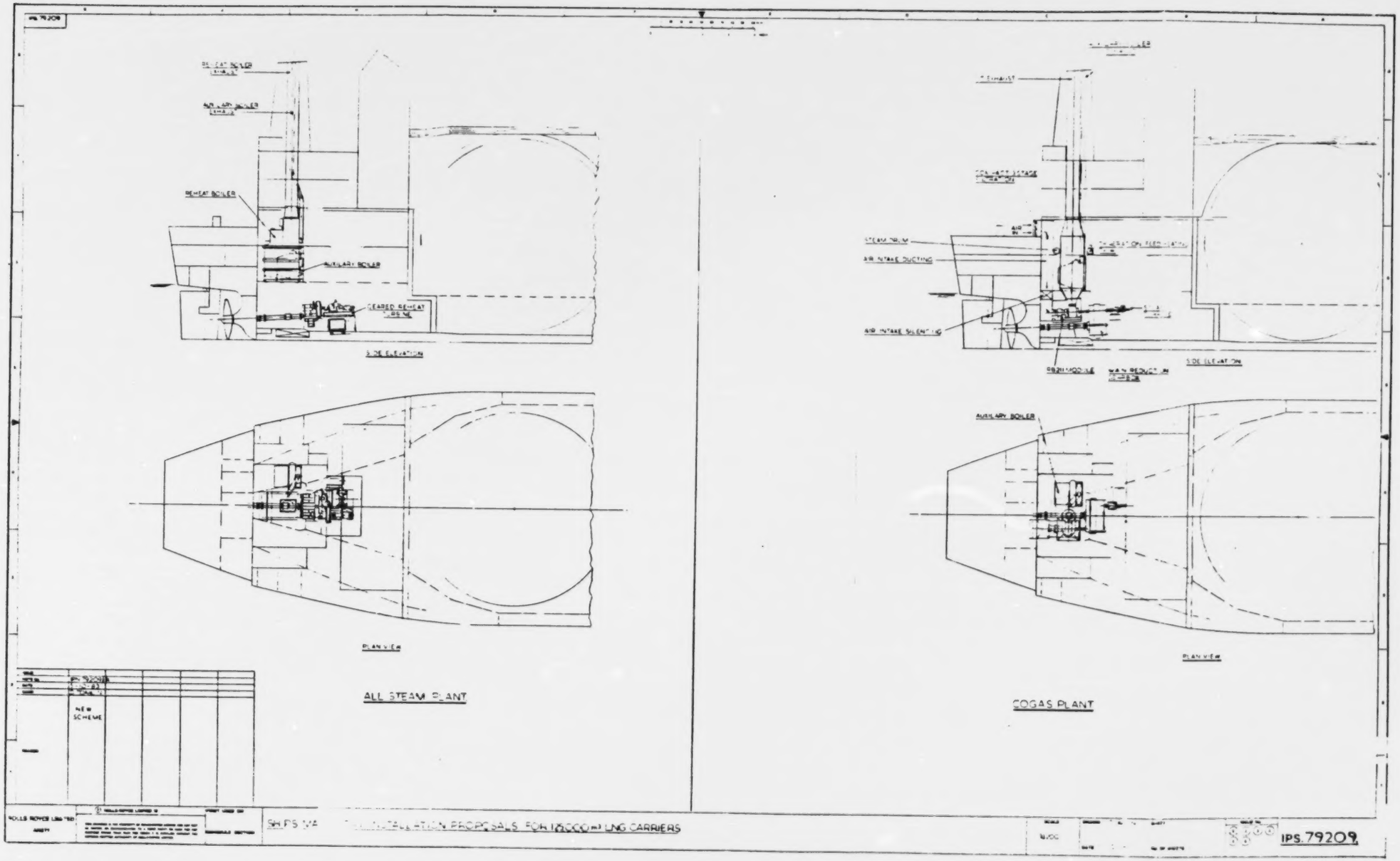


Figure 11 SHIP'S MACHINERY INSTLLATION
(COGAS AND 'ALL-STEAM' PLANTS)

- WHRB
- Auxiliary boiler
- Propulsion steam turbine
- Electrical power requirements
- Main reduction gearbox (not part of submission/study)
- Additional information

RB.211/RT.56 Module

The Rolls-Royce Limited RB.211 gas generator is coupled to a Cooper Energy Services RT.56 power turbine, both housed in an insulated enclosure, providing acoustic and heat insulation protection. Access door and inspection panels are incorporated. Ventilation of the module is by forced draught. The module can be mounted either flexibly or solidly.

The RB.211 gas generator exhaust gases drive the RT.56 power turbine which rotates the propeller shaft via the main reduction gearbox.

The exhaust gases after leaving the power turbine are directed through a WHRB to provide steam to ship's services and to a propulsion steam turbine.

Air Intake System

a) The filter housing is mounted on the main deck as shown and incorporates the following:

   - Access panel on the roof to facilitate gas generator removal.

   - A compact 3 stage air inlet filtration system which is sized for a through velocity of 7 to 9 m/sec.

   - Emergency bypass doors (if filters block-up) if required - not shown on drawing.

b) The air intake silencing section is designed for a through velocity of 30 m/sec and the length is subject to noise level required by customer.

c) The air leaves the silencing section and enters the plenum chamber thence to the gas generator intake flare.

d) Shipbuilder's supply - filter housing, air intake silencing, trunking and plenum chamber.

Exhaust System

A diffuser is fitted to the module exhaust volute bellows to change the gas path from rectangular to circular, designed to give as smooth a gas flow as possible to reduce losses. The diffuser is to be totally supported by the ship's structure to take its weight

and allow for movements and thermal expansion.

WHRB

The RB.211/RT56 module exhaust gases are directed through the WHRB, heating the water in the tubes to produce steam for:

a) propulsion steam turbine
b) ship's steam services
c) steam turbine driven compressor - if compressor is not elecrically driven (supplying pressurised LNG to RB.211 generator)

The WHRB has the following items mounted separately

- steam drum
- deaeration feed heating tank

Auxiliary boiler

a) To provide cargo handling and ship's services in port.

b) Used in emergency condition to provide 'take home' power and services in the event of a failure of the WHRB and/or gas turbine.

Steam Turbine

Receives superheated steam from the WHRB and augments the output of the gas turbine by rotating the propeller shaft via the main reduction gearbox.

Electrical Power Requirements

a) Services power generation can be supplied by either a shaft generator system driven from the main reduction gearbox backed up by an attached steam turbine or by a single steam turbine generator (second alternative not shown).

b) Standby diesel generators for bow thruster.

c) Electric motor to drive LNG compressor if steam turbine system is not used (not shown on drawing).

Main Reduction Gearbox

This is not to be part of the submission but is drawn in to show how system works as a whole and for space requirements.

Additional Information

The COGAS LNG carrier is propelled by a CPP.

2.4 'All-Steam' Plant Design - Preliminary Considerations

The selection of a suitable 'all-steam' propulsion plant for direct comparison with the COGAS plant considers principally a large main boiler with reheat with provision for an auxiliary boiler.

The reheat installation was considered as the main option because of the total plant improvement performance, in the order of 10-14% in thermal efficiency, over the non-reheat system.

The steam plant design considers all the manual requirements and design practices common to LNG ship propulsion plants including regenerative feedwater heating and freshwater distillation. A schematic of this system can be seen in Fig. 12.

The main difference between the COGAS WHRB and the 'all-steam' plant boiler is that the heat exchanger of the latter has a sophisticated type of furnace which uses a high proportion of radiant heat transfer compared with the WHRB which uses mainly convective heat transfer.

The principal competition for the main boiler is Foster Wheeler ESRD design which produces an evaporation of 65 to 76 T/h. This is similar in design to the reheat plants fitted in the 'Energy' class of oil product carriers built by Sasebo Heavy Industries Co. Ltd. This boiler is complete with superheater and superheat control, attemperator, secondary superheater, reheater with reheat control dampers, bare tube and extended surface economisers and rotary gas air heater. The boiler overall dimensions are 12,7 m high with a cross section of 8,3 m by 5,3 m, whereas the dimensions of the Foster Wheeler WHRB are 11,05 m x 5,55 m x 3,05 m respectively.

The boiler design for large tankers and carriers is dictated by the installation space available which for the vessels considered, is more accommodating in height than in width and length. Therefore, a greater furnace volume is achieved by increasing the height of the boiler. The increased height allows the burners to be positioned in the roof of the furnace where they are fired downwards producing a long flame path. The fuel output per burner is high due to the fact that the modern trend in marine steam generators is towards boilers having high steam outputs combined with the desirability of reducing the number of oil burners to a minimum to achieve simplicity of operation.

2.4.1 Reheat Boiler

These boilers are able to operate over their full range on fuel oil alone or with up to 90% heat input from natural gas boil-off.

The emissivity of the natural gas flame is normally very much less than that of the oil flame, consequently steam temperatures on gas firing are higher than for oil firing. This results from higher flue gas temperatures across the superheater which occur on gas firing due to the lower furnace heat absorption rate. Consequently

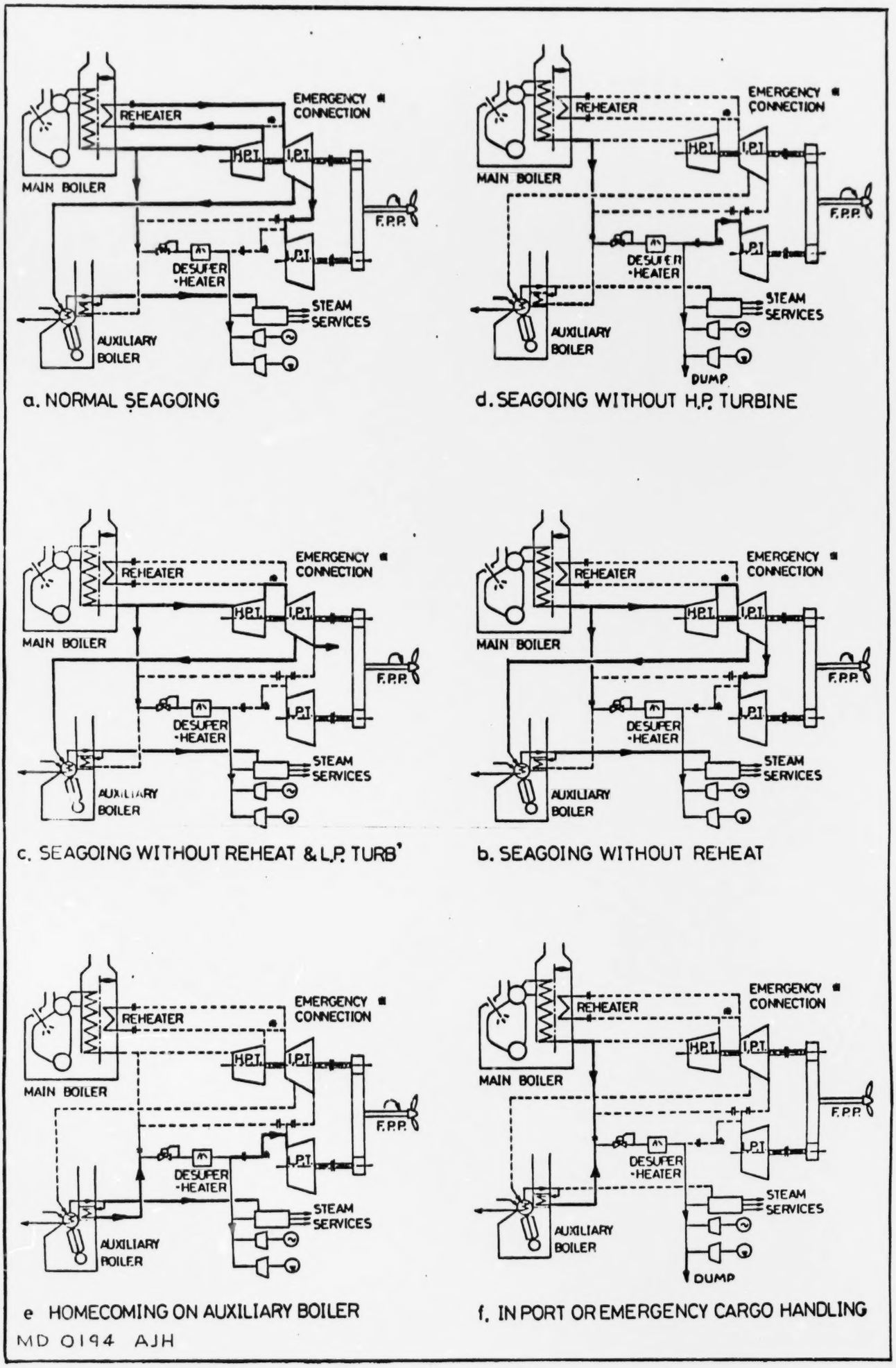


Figure 12 SCHEMATIC - 'ALL-STEAM' PLANT

both the superheater and the superheat control must be specially designed for this.

Another important point with gas fired boiler design is that for safety reasons the boilers have always been double cased to prevent the possibility of natural gas leakage into the machinery spaces.

The Foster Wheeler external superheater reheat 'D' type (ESRD) boiler is currently in service on nine very large crude oil carriers since 1977.

The design has a conventional roof fired furnace but the superheater and reheater sections have a water cooled division wall within the boiler.

Superheat control is achieved by drum on spray type attemperation whilst reheat control is achieved with partial gas by-passing of the reheater by modulation of the damper above the bypass economiser.

When the ship is manoeuvring astern there is no steam flow to the reheater. For protection, the flue damper above the reheater is closed so that the flue gases bypass the reheater. Further protection is afforded by air cooling, which is achieved by opening up a damper which allows cool air to pass through the reheater.

2.4.2 Auxiliary Boiler

The auxiliary two drum or 'D' type boiler is provided to supply 'take-home' power via the low pressure turbine and steam services in the event of a failure of the main boiler and to provide 'in-harbour' power requirements. This is of welded construction incorporating water cooling on all external walls with longitudinal tubes which are welded fin to fin. This is insulated with low temperature insulation and protected on the outside by aluminium cladding. The boiler is designed for an evaporation rate of 30 T/h at a pressure of 6200 kPa and has a dual fuel oil/gas firing capability under forced draught conditions.

2.4.3 The Reheat Steam Turbine

The steam turbine for the reheat system is a three cylinder arrangement with HP and IP cylinders coupled together running at 10 000 rpm with a separate LP cylinder running at 4400 rpm. The total power for the steam turbine arrangement being steam enters the HP cylinder at 10 000 kPa 538 deg.C. and exhausts to the reheater where its temperature is raised again to 538 deg.C. This steam then enters the IP cylinder at a pressure of 2400 kPa. It exhausts from the IP cylinder via a balanced cross over pipe into the LP cylinder, from which it flows into an underslung condenser at a vacuum of 762 mm Hg. The astern turbine is incorporated into the LP cylinder.

2.4.4 Operation of Plant

The various modes of operation for the steam plant including failure cases are as follows:

Case	Plant Condition
a) Normal seagoing	HPT, reheater, IPT, LPT, ST generator in operation. Exhaust to condenser.
b) Seagoing without using reheater	HPT, direct connection to IPT, LPT, ST generator, in operation. Exhaust to condenser.
c) Seagoing without using reheater and having an LP turbine failure	HPT, direct connection to IPT, ST generator, in operation. Exhaust to condenser.
d) Failure of the HP turbine	LPT, ST generator only in operation, fed directly from steam main. Excess steam dumped to condenser via de-superheater.
e) Home coming on the auxiliary boiler	HPT, ST generator, only in operation, fed from auxiliary boiler. Exhaust to condenser.
f) In port or emergency cargo handling	ST generator only in operation from main and auxiliary boiler. Excess steam dumped to condenser via de-superheater

where: HPT = High Pressure Turbine
IPT = Intermediate Pressure Turbine
LPT = Low Pressure Turbine
ST = Steam Turbine

2.4.5 Steam Plant Performance

The latest standard of modern reheat boilers incorporate preheaters and are fed by sophisticated feed heating systems. Plants

have shown progressive improvement in the 'all purpose fuel rate' since 1950 by the embodiment of improved components and the increase in operating pressures and temperatures. An example is the MST14 plant in 1967 which operated at 10 000 kPa and 510 deg.C., with reheat, 5 feed heaters and a gas to air heater, which gave an all purpose specific fuel consumption of 251 kg/MW h (12,6 MW). This technology is now well tried and tested.

Since then designs operating at 12 400 kPa, 538 deg.C at 237 kg/MW h (37,3 MW) and 16 500 kPa, 565 deg.C. 228 kg/MW h (37,3 MW) have become available.

LNG carriers burning boil-off gas can achieve fuel rates of 202 kg/MW h [1].

A steam turbine plant provides a highly cost effective means of propulsion with a capability for long term operation between overhaul. It has the flexibility and reliability of power production with single/dual fuel firing capability.

The 'all-steam' plant occupies slightly more machinery space than the COGAS plant, though there is slightly less space occupied by intake and exhaust ducts.

The 'all-steam' plant performance is not affected by high ambient temperatures.

Maximum power running for extended periods does not adversely affect the life of a steam plant.

Steam plant has the ability to burn a wide range of fuels including residual fuels.

2.4.6 Description of 'All-Steam' Installation Drawing (Fig 11)

The installation consists of the following:

- Main boiler
- Auxiliary boiler
- Propulsion steam turbine
- Electrical power requirements
- Main reduction gearbox

Main Boiler

The dual fuel (gas and/or heavy fuel oil) reheat/non reheat boiler, heats the water in the tubes to produce steam for:

a) propulsion steam turbine
b) ship's steam services

Auxiliary Boiler

In the event of the main boiler failing, the auxiliary boiler

will be a back-up and provide 'take-home' power and ship's services.

Propulsion Steam Turbine

The steam turbine receives superheated steam from the main boiler, or in emergency condition, from the auxiliary boiler. The geared steam turbine plant, whether reheat HP, IP, and LP turbines or non reheat HP and LP turbines, is to drive through a reversing reduction gear (the ship's propeller is of the fixed pitch type).

Electrical Power Requirements (not shown on drawing)

a) Services power generation can be supplied by either a constant frequency shaft generation system attached to the steam turbine reduction gear backed up by a separate auxiliary steam turbo generator or by a conventional multistage turbine generator fed by main steam and backed up by diesel generators.

b) Electric motors to drive LNG compressor if steam turbine system is not used.

Main Reduction Gearbox

This is not to be part of the submission, but is drawn in to show how system works as a whole and for space requirements.

2.5 Conclusions (Phase 1)

The ship's requirements, including the gas turbine power setting on the basis of the latest plant available, were sufficiently well established at this stage in order to formulate the initial design of propulsion plant.

The co-operative suppliers of both COGAS and 'all-steam' plant were identified and are as follows:

Foster Wheeler Power Products Limited

NEI International Combustion (Cochran Limited)

Sunrod A.B. Sweden

Senior Green Limited

Bigalow Liptak of Canada Limited

Westinghouse Canada Inc.

GEC Turbine Generators Limited

Kawasaki Heavy Industries Limited

AEG Telefunken

Initial feelings, in discussion with the manufacturers, favoured the more simple and proven solution of a single pressure turbine design for the waste heat system rather than a more complex dual pressure steam turbine and boiler arrangement. All the potential suppliers agreed with the view of Rolls-Royce that the simple and proven single pressure waste heat recovery steam system was likely to be the preferred choice. A dual pressure system would involve significant extra capital costs and would result in a very much more complex plant for a marginal improvement in thermal efficiency. However, the suppliers were invited to consider the option of multi-pressure systems when responding to the Engineering Enquiry Specification (see Section 4.1. part (a)). In the event, they all responded with single pressure systems.

3. PRELIMINARY COGAS OPTIMIZATION (Study Phase 2)

3.1 General

The purpose of this phase of the study was to identify the optimum gas turbine design point power setting to achieve the maximum plant efficiency within practical bounds.

The defined system concept was:

- simple single pressure WHRB and steam turbine system
- services' steam via desuperheater
- services' power via reduction gear driven shaft generation arrangement.

The boiler and steam turbine assumed characteristics for the purposes of the combined cycle simulation were as follows:

Boiler stop valve pressure (BSVP)	2600 kPa
Superheater approach temperature difference	40 deg.C.
Evaporator pinch temperature difference	20 deg.C.
Boiler radiation losses	1,5%
Drum blow down flow	1,5%
Superheater pressure drop	1,9%
Steam main pressure drop	100 kPa
Steam main temperature drop	5 deg.C.
Turbine stop valve pressure drop	5%
Steam turbine blading efficiency	82%
Steam turbine mechanical and gland efficiency	97,5%
Steam turbine reduction gear efficiency	97,5%
Steam services bleed via desuperheater:	
mass flow	4 T/h
temperature	340 deg.C.
pressure	1850 kPa
Feedwater de-aerator pressure	270 kPa

L.P. de-aerator boiling loop pinch	25 deg.C.
Condenser pressure	5,1 kPa

Using these fixed characteristics, calculations were carried out to determine the gas turbine setting which would satisfy the total 'at sea' power requirements of 21 404 kW in addition to the services steam flow of 4T/h for cofferdam heating.

By working through the calculations using a range of gas turbine settings it was possible to satisfy the 'at sea' requirements of 16 816 kW at the RB.211/RT56 output flange with an ambient operating temperature of 9 deg.C. By performing a whole suite of calculations and altering the evaporator pinch temperature for increasing values of boiler operating pressure it was possible to look more objectively at the design point parameters that had been chosen initially for the steam system - see Figs. 13, 14, and 15.

Figure 13 - This shows how the power output of the steam turbine is affected by the combination of evaporator pinch and boiler stop valve pressure.

Figure 14 - This diagram illustrates the effect on boiler tube area of changing the combination of evaporator pinch and boiler stop valve pressure.

Figure 15 - This diagram illustrates the effect on the power per boiler area ratio when the evaporator pinch and boiler stop valve pressure combination is changed.

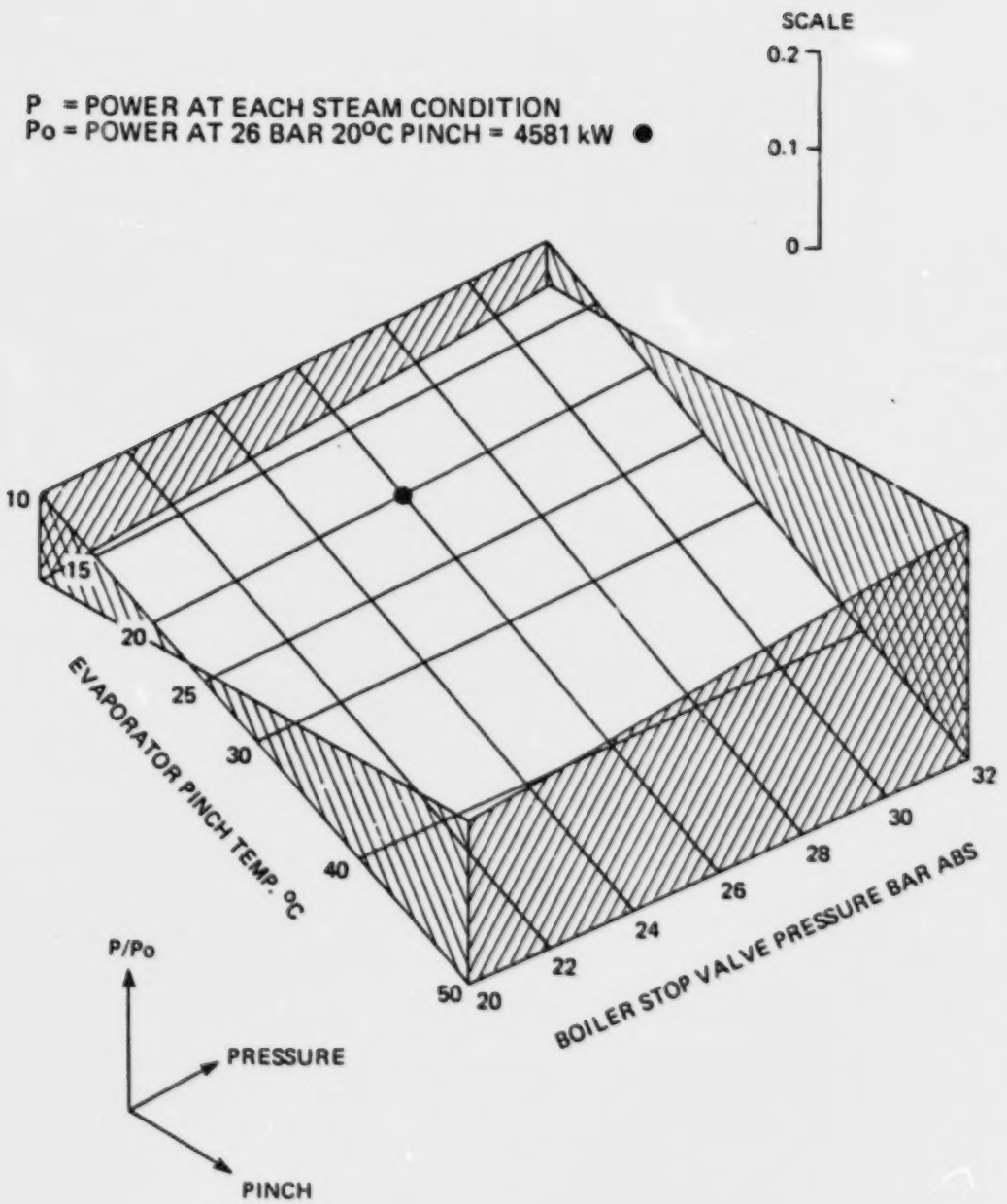
A reduction in boiler operating pressure increases boiler efficiency but also increases the size of the steam turbine. An increase in boiler area not only increases the boiler efficiency but also increases the size of the boiler.

It was recognised at this stage that in order to realise maximum efficiency of the total system the arrangement would need to operate in a 'sliding pressure' manner, where the boiler pressure follows the rate of steam flow. In order to accommodate this type of operation the 2600 kPa boiler stop valve pressure setting was firmly adopted as the most suitable design level of pressure for steam generation. Secondly, the engine room space potential was such that a boiler having the maximum possible heat recovery could be fitted, ie., one with an evaporator pinch temperature of around 20 deg.C.

Increased Ratings

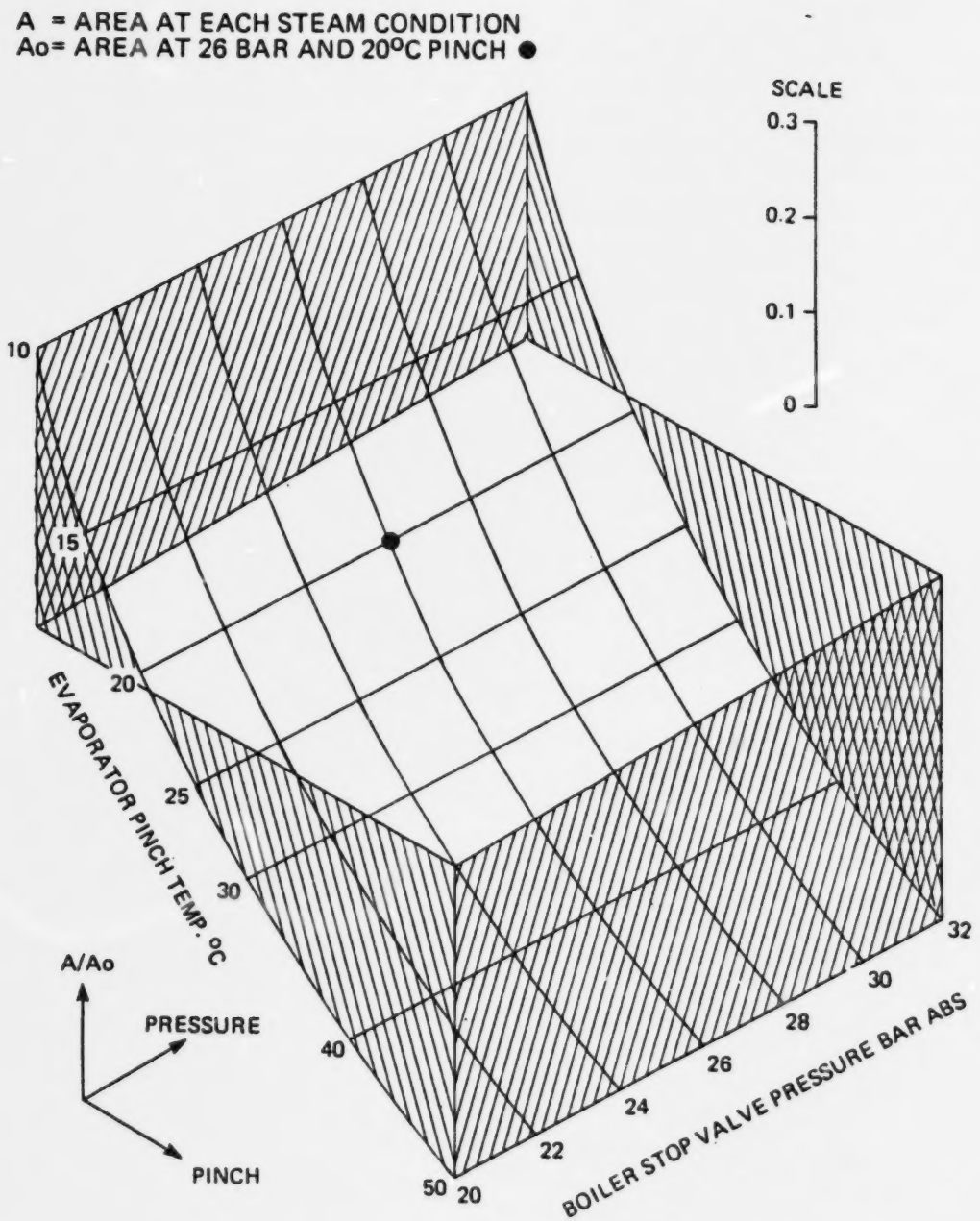
In the event of the WHRB being inoperative, then the bypass duct is used so that the gas turbine can be taken up to its Maximum Continuous Rating of 25 079 kW at 9 deg.C.

The WHRB is designed to operate independantly of the gas



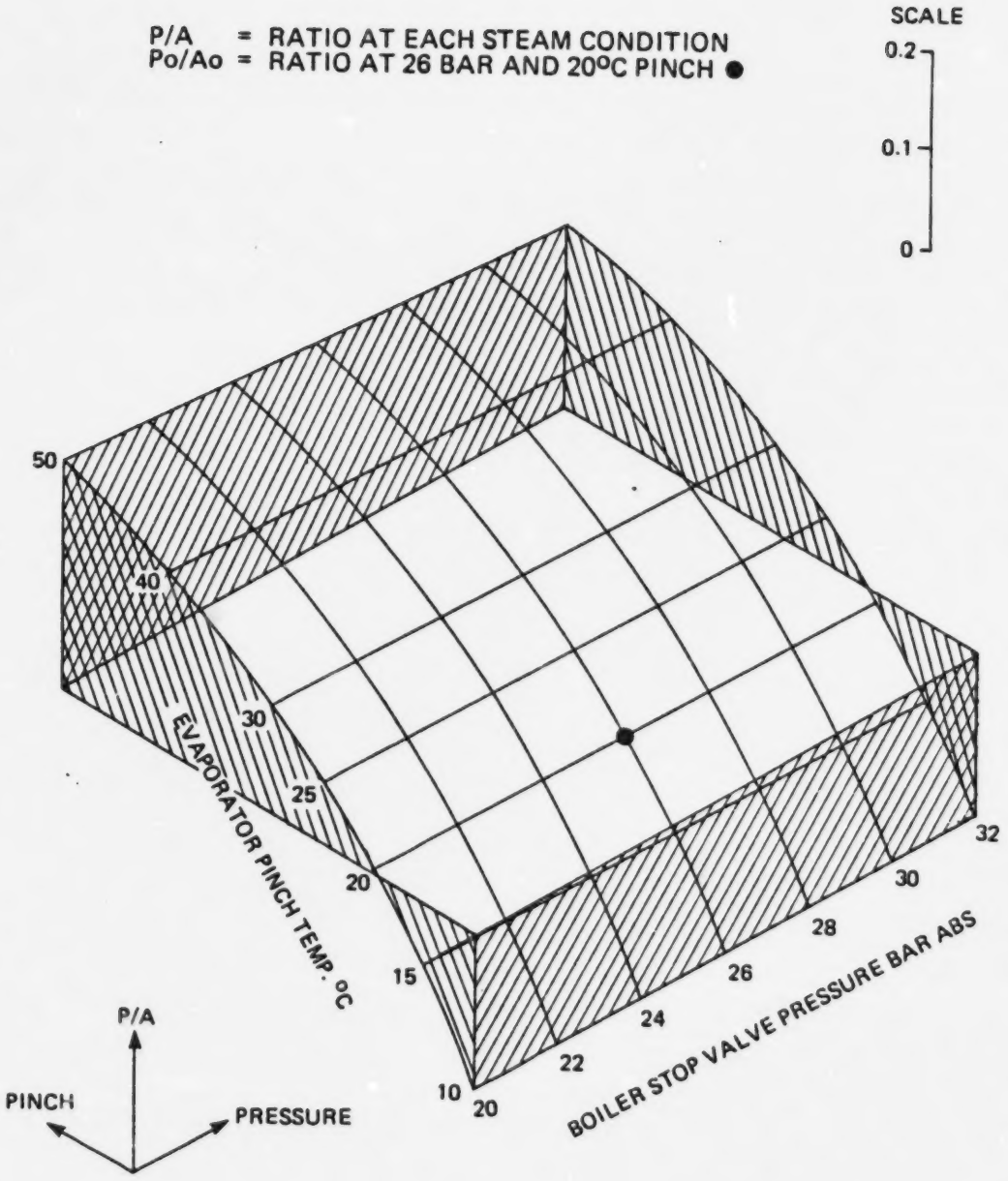
BOILER DESIGN POINT BASED ON 4581 kW STEAM TURBINE OUTPUT IN ADDITION TO 4T/HR SERVICE STEAM AT 18.5 BAR, 340°C, 9°C AIT

Figure 13 BOILER PERFORMANCE OPTIMISATION
STEAM TURBINE OUTPUT POWER RATIO (P/PO)
VERSUS STEAM PRESSURE AND BOILER PINCH



BOILER DESIGN POINT BASED ON 4581 kW STEAM TURBINE OUTPUT IN ADDITION TO 4 T/HR SERVICE STEAM AT 18.5 BAR, 340°C, 9°C AIT

Figure 14 BOILER PERFORMANCE OPTIMISATION
BOILER TUBE AREA RATIO (A/AO)
VERSUS STEAM PRESSURE AND BOILER PINCH



BOILER DESIGN POINT BASED ON 4581 kW STEAM TURBINE OUTPUT IN ADDITION TO 4T/HR SERVICE STEAM AT 18.5 BAR, 340°C, 9°C AIT

Figure 15 BOILER PERFORMANCE OPTIMISATION BOILER TURBINE OUTPUT POWER/BOILER AREA(P/A)VERSUS STEAM PRESSURE AND BOILER PINCH.

turbine, 'refiring' on gas fuel using its own set of burners. To maintain a 'take-home' speed of 11 knots in the event of gas turbine shutdown the WHRB will be capable of operating at an equivalent Maximum Continuous Rating in the refiring mode. This will produce an estimated steam delivery pressure in the order of 3900 kPa and a correspondingly higher steam turbine output power.

Extra LNG required at any time will be from forced vaporization and maintenance of services' steam supply will be met by the 4T/h auxiliary boiler using diesel fuel if required.

3.2 Simulation Steam Plant Design

The waste heat boiler used in the simulation employs convective heat transfer with an air to fuel ratio in the region of 60 and greater as opposed to 15 or so for ordinary fired boilers. The single pressure design includes a steam separating drum and de-aerating feed heating system similar in concept to that used widely in motor ships. In this instance however, a low pressure boiling loop is incorporated to supply a de-aerating feed heater and the addition of an auxiliary gas burner system provides a means of firing the boiler without using the gas turbine.

In addition to providing main superheat high pressure steam for the steam turbine a de-superheated bleed is taken to supply the ship's services' steam requirement (4T/h, 1850 kPa, 340 deg.C.).

A simulation study based on a system using desuperheated H.P. steam as outlined above was embarked upon using a comprehensive and detailed combined cycle computer program developed by Rolls-Royce Limited especially for the study of waste heat boiler/steam turbine design interaction.

The alternative to taking the high pressure bleed for services would be to introduce a dedicated evaporator into the stack design. This would yield a more efficient plant but at the same time increase the complexity of the waste heat boiler and probably the cost.

As a basis on which to write an initial specification (Enquiry Document, Rolls-Royce Report No. MD.027) and to arrive at a suitable gas turbine power setting it was decided, in view of the timescale, to go ahead with the desuperheat design in order to give the various boiler and steam turbine manufacturers time to respond.

Other Design Features

Careful consideration was given to the temperature of the feedwater in the dedicated stack heating tubes so as to avoid corrosion from precipitated suphates when the gas turbine is run on distillate fuel. The dew point of the exhaust from the RB.211 at the design running power of 16 816 kW is calculated to be 28 deg.C.

When the waste heat boiler is being refired and the gas turbine

is shut down ('take-home' condition) a damper and forced draught fan will be brought into play. The damper will prevent re-firing combustion gases from re-entering the gas turbine while the fan will provide the necessary dilution air to the dedicated burner system.

3.3 Conclusions

The design used in the simulation exercise demonstrates the maximum efficiency of operation that can be achieved within practical limits of boiler size and cost. The system is simple, flexible in operation, has good part power efficiency, is moderately rated and as such will have a long life.

4. ENGINEERING ENQUIRY SPECIFICATION PRODUCTION, ISSUE AND RESPONSE ASSESSMENT (Study Phase 3)

4.1 General

This phase identifies the engineering requirements of a practical propulsion plant in the form of outline specifications to cover the major items of the COGAS and 'all-steam' reheat installations. These specifications provided the basis against which equipment manufacturers were invited to quote for the provision of competitive plants in the two cases.

The technical requirements were based upon known design practice, eg.,

- previous combined cycle work carried out by Rolls-Royce for studies and proposals
- established marine steam propulsion expertise of the major suppliers and their recommendations
- recommendations of other feasibility studies, eg.
  MARAD - George C. Sharp Inc.
  SOLAR - USN design contract
- results of preliminary investigations and optimisations in the foregoing phases
- COGAS optimisation in Phase 2
- Reheat investigations in Phase 1

The specifications were written on the basis of plant designs estimated to be realistic within the timescale. They were intended to provide design guidance but also to allow opportunity for suppliers to recommend solutions so that the balance could be maintained between practicality and the best possible plant performance.

Recommendations were invited from manufacturers with regard to:

a) COGAS

- boiler size
- choice of single or multipressure system
- burner arrangement for auxiliary firing operation
- amount of services' steam production and type of boiler
- ducting bypass solutions for gas turbine/waste heat boiler

- choice of forced draft or natural circulation waste heat boiler

- auxiliary services' power and electrical generation methods

- safety of operation

The desuperheated services' steam option was intended for guidance only and did not constrain suppliers to necessarily adopt this method themselves.

b) Reheat

- regenerative economy measures

- fuelling arrangements

- auxiliary steam production for emergency operation of main steam turbine

- safety of operation

The importance of Canadian content was stressed for both COGAS and Reheat systems.

4.2 Engineering Enquiry Specifications

The complete engineering enquiry specification which was sent out to suppliers included the following:

- projected machinery configurations for both plants

- machinery classification requirements

- propulsion and auxiliary plant power requirements

- propulsion system operating conditions

- environmental requirements

- fuel specification

- maintenance actions

- scope and description of supply to be quoted for both COGAS and 'all-steam' plant

All suppliers who were willing to co-operate in the assembly of data were provided with the full enquiry document together with supporting gas turbine and ship layout information

4.3 Manufacturers' Responses

Canadian suppliers showed generally only small interest in the

study and large established companies such as Babcock & Wilcox (U.K.) and Stal Laval declined involvement on all aspects.

The only Canadian responses were from:

Westinghouse (Canada)

Babcock & Wilcox (Canada)

Bigelow - Liptak

4.3.1 The COGAS Plant

a) WHRB

Companies who indicated willingness to supply information on the above were:

Foster Wheeler Power Products

NEI International Combustion (Cochran) Limited

Aalborg-Vaerft A/S Denmark

E. Green and Sons Limited

Sunrod A.B. Sweden

Bigelow Liptak of Canada Limited

Kawasaki Heavy Industries

Babcock and Wilcox Canada

NEI International Combustion (Cochran) Limited informed us of their lack of resources and passed the work to NEI Nuclear Systems Limited.

- Aalborg Boilers, although keen initially, did not respond to the enquiry document
- E. Green & Son (now Senior Green Limited) carried out considerable study work upon receipt of the document
- Sunrod AB made early technical responses
- Bigelow Liptak with a Deltak design eventually declined to participate in the study as a marine application
- KHI, although keen at first, did not respond finally
- Babcock & Wilcox (Canada) Limited could only assist with standard reheat boiler plant.

The preliminary WHRB was a single pressure unit with desuperheated services and the initial liason centred around this particular design.

- NEI responded quickly with outline ideas on this system

- Foster Wheeler provided full design and specification on this very early on

Sunrod responded by suggesting an alternative and more economical method of services' steam generation via a dedicated LP generator built into the boiler. Sunrod also expressed concern over the effect on feed pump requirement of floating pressure operation, especially at the 'take-home' condition where bypassing might be needed to maintain moderate system pressures.

Sunrod and Senior Green have incorporated a deaerator heater, positioned at the top of the WHRB, for the sole purpose of heating the feed water in the deaerator. However, Foster Wheeler and NEI propose to use some of the steam generated in the LP system for carrying out this task. The latter proposal is preferred because it is simpler and less expensive.

Only the Foster Wheeler system has dual fuel burners which would allow the WHRB to run on marine diesel or some other liquid fuel as well as vapourised natural gas. This would provide the LNG carrier with an emergency 'take-home' facility should there be any problems with the primary natural gas supply, which will be natural boil-off or forced vapourisation from the ship's LNG cargo.

No concern was expressed over feedheating temperatures.

The concept of a fast response 'once-through' WHRB was not considered in the Enquiry Document although, in view of current developments for warship applications by the US Navy [2], it was discussed informally with the boiler manufacturers. They were free to propose this type of design but all decided against it on the grounds of cost and degree of risk. They considered that adequate speed of response could be achieved with conventional drum boilers.

No major queries emerged as a result of issuing the enquiry document except for the changes to the services' steam arrangement as already mentioned.

The Sunrod suggestion made much better use of the available gas turbine exhaust energy resulting in overproduction of HP steam for the ship power requirements. This caused the design point for the gas turbine to be reduced from 16 816 kW to 16 331 kW, with a corresponding increase in the overall efficiency of the plant over the small boiler arrangement.

The new design point requirements were sent out to those WHRB suppliers willing to reconsider the design. These suppliers were:

Sunrod A/B Sweden

Foster Wheeler Power Products

Senior Green Limited

The main features of the submissions from the various suppliers are shown in Figs 16 and 17, (Section 4.4).

b) Waste Heat Propulsion Steam Turbine and Condenser

Preliminary design work using the desuperheated services steam approach identified a COGAS steam turbine shaft power output of 4581 kW to meet both propulsion and services electrical power (shaft driven generator). Although an alternative arrangement using two separate steam turbines was put forward in the enquiry document, GEC Turbine Generators and Westinghouse both agreed that a single large turbine with a shaft generator system and back-up turbine would be the most cost effective solution.

Recalculation of the design point increased the steam turbine output required to 5262 kW but this did not present any problems, as the same turbine frame could be used.

The main features of the submissions from the propulsion steam turbine suppliers are shown in Fig. 18, (Section 4.4).

The four steam turbine manufacturers approached were:

GEC Turbine Generators Limited

Westinghouse (Canada) Inc.

De Laval Turbine (Canada) Limited

Kawasaki Heavy Industries

De Laval and KHI did not provide a response

c) Auxiliary boiler for COGAS plant

The initial requirement for a 3UT/h auxiliary boiler was now considered unnecessary because of the superior level of redundancy offered within this type of system over the 'all-steam' plant. There was, however, still a requirement for a single 4T/h water tube boiler to maintain steam services when the gas turbine and WHRB are shut down.

NEI, Sunrod, and Greens quoted for this type of boiler

d) Power Generation and Auxiliary Turbine Driver Systems

It was pointed out that the provision of a single or twin-stage auxiliary turbine driver, although a cheap solution, would consume a

great deal of steam. The specification was modified to invite proposals for more efficient multi-stage units.

At the same time, it was decided to increase the LNG compression power from 400 kW (preliminary estimate) to 600 kW (specified by manufacturers in response to the Enquiry Document).

4.3.2 The 'All-Steam' Plant

The enquiry document invited responses covering either a reheat or non-reheat 'all-steam' plant. All parties recommended a reheat main boiler and turbine with a non reheat auxiliary boiler as back up.

a) Reheat boiler

The following companies agreed to submit boiler information:

Foster Wheeler Power Products Limited

NEI Nuclear Systems via their sister company NEI International Combustion

Sunrod AB Sweden

Babcock & Wilcox (Canada) Limited

Foster Wheeler responded with substantial descriptive and technical information on both boilers, of their own design.

Sunrod AB offered equipment to a combustion engineering design using their own air preheater.

NEI and Babcock & Wilcox did not respond.

No problems were encountered as regards dual fuel capability. The units are standard designs which although not as advanced as the design prospects for the future are equivalent to the latest standard of unit in service.

The main features of the submissions from the two suppliers are given in Figs. 19 and 20, (Section 4.4).

b) 'All-Steam' plant propulsion turbine and condenser

The following steam turbine manufacturers agreed to co-operate:

GEC Turbine Generators Limited

Westinghouse (Canada) Inc.

De Laval Turbine (Canada) Limited

Kawasaki Heavy Industries

Westinghouse stated that the smallest reheat turbine that they could offer from their range was 44 742 kW. Alternatively, they could offer a smaller reheat design subject to suitable support on engineering design costs or perhaps build a GEC design under licence. Finally, it was agreed they should offer a conventional non-reheat turbine design.

GEC responded with full descriptive and technical coverage for a reheat turbine of the required size.

Westinghouse did not respond with the non-reheat turbine in time for it to be included in the study.

The main features of the GEC submission are shown in Fig. 21, (Section 4.4).

c) Power Generation and auxiliary turbine driver system for the 'all-steam' plant

The same manufacturers as above were approached but only Westinghouse was able to respond on the multi-stage turbines for electrical generation. The electrical requirement for this type of plant is slightly less than the COGAS due to the smaller feed pump duty of 347 kW.

4.4 Tabulated Comparisons Between the Respondings Suppliers Proposals for both the COGAS and Steam Plants

Note: Comparisons between both the auxiliary boilers and the steam turbine generator systems were not possible because insufficient information was available.

	FOSTER WHEELER W.H.R.B.	GREEN W.H.R.B.	SUNROD W.H.R.B.	NEI W.H.R.B.
Approximate Dimensions of W.H.R.B. (m)	11,05 x 5,55 x 3,05	5,344 x 7,05 x 3,79	9,268 x 7,65 x 5,75	7,4 x 5,5 x 5,5
Minimum Tube Withdrawal Space (m)	5,5	6,1	6,0	Not required as tube failures be blanked off.
Weight of W.H.R.B. (tonne) (incl. water)		150	158	112 (excl. water)
W.H.R.B. Heating Surface Area (m^2)	8850	12 177	7606	4758
Tube Material	LRS Grade 320 Carbon Steel	BS3059/620 and BS3059/440	ST.35.8/II according DIN 17175 Seamless	Carbon steel SA 179 or SA 192
Construction of W.H.R.B.	Welded	Welded	Welded	Welded
L.P. Drum				
Material	Carbon steel, Lloyds Grade 460	BS1501-151-430A		
Length (m)	3,2	3,5	3,4	
Diameter (m)	1,07	1,2	1,2	N/A

Figure 16 COMPARISON OF W.H.R.B. SYSTEMS

	FOSTER WHEELER W.H.R.B.	GREEN W.H.R.B.	SUNROD W.H.R.B.	NEI W.H.R.B.
H.P. Drum				
Material	Carbon steel, Lloyds Grade 460	BS1501-151--430A		SA 151 or SA 156
Length (m)	3,2	3,5	4,6	3,0
Diameter (m)	1,07	1,2	1,6	1,5
By-pass or Dry-firing Option	Both available		By-pass	Both available
Refiring option	Dual fuel burner system	Gas fuelled linear duct burner system	Gas fired duct burner system	

Figure 16 COMPARISON OF W.H.R.B. SYSTEMS (Cont'd)

Figure 16 COMPARISON OF W.H.R.B. SYSTEMS (Cont'd)

	FOSTER WHEELER WHRB SYSTEM AND GREEN'S AUX. BOILER	GREEN'S WHRB SYSTEM AND AUX. BOILER	SUNROD WHRB SYSTEM AND GREEN'S AUX. BOILER	NEI. WHRB AND GREEN'S AUX. BOILER
Design Load :-				
A.I.T. (°C)	9	9	9	9
Exhaust gas flow (kg/h)	282 168	282 150	285 272	285 264
Exhaust gas temp at WHRB inlet (°C)	417	417	420	420
Gas turbine power (kW)	16 331	16 331	16 816	16 816
HP steam flow (kg/h)	22 874	22 900	24 000	19 914
HP steam temp at outlet (°C)	380	385	380	380
HP steam pressure at outlet (kPa)	2600	2600	-	2600
LP steam flow (kg/h)	-	4000	4000	10 760
Exhaust gas temp at WHRB outlet (°C)	162,5	153	159	-

Figure 16 COMPARISON OF W.H.R.B. SYSTEMS (Cont'd)

	FOSTER WHEELER WHRB SYSTEM AND GREEN'S AUX. BOILER	GREEN'S WHRB SYSTEM AND AUX. BOILER	SUNROD WHRB SYSTEM AND GREEN'S AUX. BOILER	NEI. WHRB AND GREEN'S AUX. BOILER
Take Home Condition:-				
A.I.T. (°C)	9	9	9	9
Exhaust gas flow (kg/h)	334 692	306 200	333 280	-
Exhaust gas temp at WHRB inlet (°C)	480	480	480	561
HP steam flow (kg/h)	33 674	29 850	33 300	29 750
HP steam temp at outlet (°C)	410	427	410	410 (after desuper-heating)
HP steam pressure at outlet (kPa)	3900	4000	3800	3900
LP steam flow (kg/h)	-	5800	6300	13 016
Exhaust gas temp at WHRB outlet (°C)	190	158	168	-

EXTENT OF SUPPLY	FOSTER WHEELER W.H.R.B.	GREEN'S W.H.R.B.	SUNROD W.H.R.B.	NEI W.H.R.B.
Superheater	•	•	•	Detailed information not available at present time
H.P.Generator	•	•	•	
H.P.Economiser (Stage 2)	•	•	•	
L.P.Generator	•	•	•	
H.P.Economiser (Stage 1)	•	•	•	
L.P./H.P.Economiser	•	o	o	
Deaerator Generator	o	•	•	
Deaerator	•	•	•	
L.P.Drum	•	•	•	
H.P.Drum	•	•	•	
Feed and Circulating Pumps	•	•	•	
Force Draught Fan and Driver	•	•	•	
Boiler Valves and Mountings	•	•	•	
Burners	Dual Fuel	Gas	Gas	

Figure 17 W.H.R.B. - COMPARISON OF EXTENT OF SUPPLY

Figure 17 W.H.R.B. - COMPARISON OF EXTENT OF SUPPLY (Cont'd)

EXTENT OF SUPPLY	FOSTER WHEELER W.H.R.B.	GREEN'S W.H.R.B.	SUNROD W.H.R.B.	NEI W.H.R.B.
Refiring option	•	•	•	
Boiler local instrumentation and control	•	not known	•	
Boiler and boiler pipework lagging and insulation	•	o	o	
Boiler integral pipework	•	o	•	
Ducting	o	o	Transition duct to connect boiler package to gas turbine and silencer(excluding insulation.)	
Dampers	•	not known	•	

Key:-

- • Item included in extent of supply
- o Item not included in extent of supply

	WESTINGHOUSE STEAM TURBINE	G.E.C. STEAM TURBINE
Approximate Dimensions of Steam Turbine (m)	1,73 x 2,56 x 2,00	2,44 x 3,05 x 2,87
Approximate Dimensions of Condenser Unit (m)	Not given	2,13 x 3,96 x 2,95
Materials :-		
Inlet Casing	Cr.Mo. steel	Carbon steel
Exhaust Casing	Carbon steel	Cast iron
Rotor	Alloy steel	Cr.Mo.V .
Blades	Stainless steel	12% Cr.Mo.V and 12% Cr.Mo.Ni. steels
Trips	Overspeed condition, low oil pressure, remote trip (electrical), manual trip (local).	Overspeed condition, low oil pressure, remote trip (electrical), manual trip (local), exhaust pressure high and low, turbine bearing temperature high, turbine shaft eccentricity high.
Budgetary Price (including condenser) Canadian dollars	550 000 F.O.B. Canada	814 500 Ex Works

Figure 18 COGAS PLANT - COMPARISON OF STEAM TURBINES FOR PROPULSION

Figure 18 COGAS PLANT - COMPARISON OF STEAM TURBINES FOR PROPULSION (Cont'd)

	WESTINGHOUSE STEAM TURBINE	G.E.C. STEAM TURBINE
Nominal Rated Design Point		
H.P. steam at turbine throttle valve :-		
Pressure (kPa)	2500	2500
Temperature (^{o}C)	375	372
Flow rate (kg/h)	22 860	-
Exhaust condenser pressure (mm Hga)	38	38
Rating	5262 kw at 6000 rpm	5262 kw at 8500 rpm
Take Home Condition:-		
H.P. steam at turbine throttle valve :-		
Pressure (kPa)	3790	3790
Temperature (^{o}C)	405	404,7
Flow rate (kg/h)	30 790	29 920
Exhaust condenser pressure (mm Hga)	53	53
Rating (kW)	7372	7136
Weight of steam turbine plus condenser (Tonnes)	-	26

Figure 19 STEAM PLANT - COMPARISON OF REHEAT AND AUXILIARY BOILERS

	FOSTER WHEELER	SUNROD
Approximate Dimensions(m):-		
Reheat Boiler	12,7 x 8,3 x 5,3	17,7 x 9,3 x 6,2
Auxiliary Boiler	10,7 x 4,9 x 5,8	10,0 x 8,1 x 5,1
Construction	Welded	Welded
Budgetary Price for Reheat plus Auxiliary Boilers (Canadian Dollars)	3 945 800 (ex Works)	4 615 385 (F.O.B. Karlsham, Sweden)

Figure 20 STEAM PLANT - COMPARISON OF REHEAT BOILER PERFORMANCE DATA

	FOSTER WHEELER	SUNROD
Superheater outlet pressure (kPa)	10 000	10 200
Feed water temp. (°C)	223	223
Evaporation (kg/h)	72 408	76 460
Superheater outlet temp. (°C)	538	538
Reheater outlet temp. (°C)	538	538
Reheater steam flow (kg/h)	64 400	64 400
Efficiency (%)	86	86
Gas consumption (kg/h)	4380	4045
Fuel oil consumption (kg/h)	4600	580

	G.E.C. STEAM TURBINE	WESTINGHOUSE STEAM TURBINE
Arrangement of steam turbine	3 cylinder :- H.P.cylinder) 10 000 rpm I.P.cylinder) L.P.cylinder 4400 rpm	The requirement for a 25 000 HP main propulsion steam turbine falls under the charter of the Marine Division of Westinghouse Electric Corporation, Sunnyvale, California. At the present time we have not received any information from them.
H.P.cylinder :- steam entry temp.(°C) steam entry pressure (kPa)	538 10 000	
I.P.cylinder :- steam entry temp (°C) steam entry pressure (kPa)	538 2420	
Feedheaters	2 - H.P. heaters 1 - Deaerator heater 2 - L.P. heaters	
Final feed temp.to boiler (°C)	223	
Weight of turbine and condenser (Tonnes)	60	

Figure 21 STEAM PLANT - REHEAT STEAM TURBINES FOR PROPULSION

Figure 21 STEAM PLANT - REHEAT STEAM TURBINES FOR PROPULSION (Cont'd)

	G.E.C.	WESTINGHOUSE
H.P. turbine materials:-		Information not forthcoming.
Inlet valve chests	2¼% Cr.Mo. cast	
Nozzle boxes	2¼% Cr.Mo. cast	
H.P.casing	2¼% Cr.Mo. cast	
Rotor	1% Cr.Mo.V.forging	
1st stage blades	12% Cr.Mo.V.Nb bar	
Remaining 5 stages of blades	12% Cr.Mo.V.bar	
I.P. turbine materials:-		
Nozzle boxes	2¼% Cr.Mo. cast	
I.P. casing	2¼% Cr.Mo. cast	
Rotor	1% Cr.Mo.V.forging	
1st stage blades	12% Cr.Mo.V.Nb.bar	
Remaining 4 stages of blades	12% Cr.Mo.V.bar	
L.P. turbine materials:-		
Cylinder and exhaust casing	Steel plate fabrication to B.S.S. 4360-40C	
Rotor	2½% Ni.Cr.Mo.steel forging	
1st six ahead stage blades	12% Cr.Mo.V. bar	
Last ahead stage blades	12% Cr.Ni.Mo.steel forging	
All astern blades	12% Cr.Mo.V.	

	G.E.C.	WESTINGHOUSE
Budgetary Price (including condenser) Canadian dollars.	2 715 000 Ex Works	
Additional cost for feed heaters, deaerator and interconnecting feed pipework. Canadian dollars.	1 538 500 Ex Works	

Figure 21 STEAM PLANT - REHEAT STEAM TURBINES FOR PROPULSION (Cont'd)

5. SHIP PERFORMANCE AND OPERATIONAL ASSESSMENT

5.1 General

This section deals with the performance aspects of the proposed propulsion systems. Basically there are three different designs. These are:

a) Small COGAS

b) Large COGAS

C) Reheat Steam

The first of these is the type of plant submitted by NEI. Early work carried out by Rolls-Royce identified a power setting for the RB.211-24C of 16 816 kW, which provided propulsion power as well as ship's services' steam via a spray desuperheater.

After discussions with the various recovery boiler manufacturers it was decided to alter the design to one employing a dedicated LP system for ship's services' steam. This meant that the boiler although slightly larger and more complex was more efficient at meeting the required steam demand and so the gas turbine power could be reduced.

NEI decided to meet us halfway and adopted the new LP generator design but kept the old gas turbine rating (and correspondingly small steam turbine power) yielding a slightly lower overall plant efficiency.

The remaining equipment suppliers were prepared to carry out design work on the new COGAS design employing the larger steam section and consequently smaller rating at the gas turbine. Using a larger boiler whilst still meeting the same overall power level with a smaller gas turbine rating, immediately means an increase in the overall plant efficiency. These designs of plant were termed 'large' COGAS.

The final type of propulsion plant looked at was the 'all-steam' using reheat equipment instead of a conventional 'straight through' turbine and boiler arrangement.

The submissions are as follows:

(Where appropriate a recovery boiler has been matched to the steam turbine in an attempt to maximise Canadian content, maintain a high efficiency and minimise cost).

Small COGAS - NEI International Combustion Limited WHRB /Westinghouse Canada Inc. steam turbine.

Large COGAS	- Foster Wheeler Power Products Limited WHRB /Westinghouse Canada Inc. steam turbine. - Foster Wheeler Power Products Limited WHRB /GEC Turbine Generators Limited steam turbine. - Greens WHRB/Westinghouse Canada Inc. steam turbine. - Sunrod A.B. Sweden WHRB/Westinghouse Canada Inc. steam turbine.
'All-Steam'	- Foster Wheeler Power Products Limited re-heat boiler/GEC reheat steam turbine.

5.2 Description of COGAS (Small and Large) Installation Drawing (Figs. 22 and 23)

The installation consists of the following:

- RB.211/RT56 module
- Air intake system
- Exhaust system
- WHRB
- Auxiliary boiler
- Propulsion steam turbine
- Steam turbine/generator pack
- Diesel generator pack
- Main reduction gearbox (not part of submission/study)
- Additional Information

5.2.1 RB.211/RT56 Module

The Rolls-Royce Limited RB.211 gas generator is coupled to a Cooper Energy Services RT56 power turbine, both housed in an insulated enclosure, providing acoustic and heat insulation protection. Access door and inspection panels are incorporated. Ventilation of the module is by forced draught. The module can be mounted either flexibly or solidly.

The RB.211 gas generator exhaust gases drive the RT56 power turbine which rotates the propeller shaft via the main reduction gearbox.

The exhaust gases after leaving the power turbine are directed through a WHRB to provide steam to ship's services and to a propulsion steam turbine which augments the output of the gas turbine. If the WHRB is inoperative, then the exhaust gases are

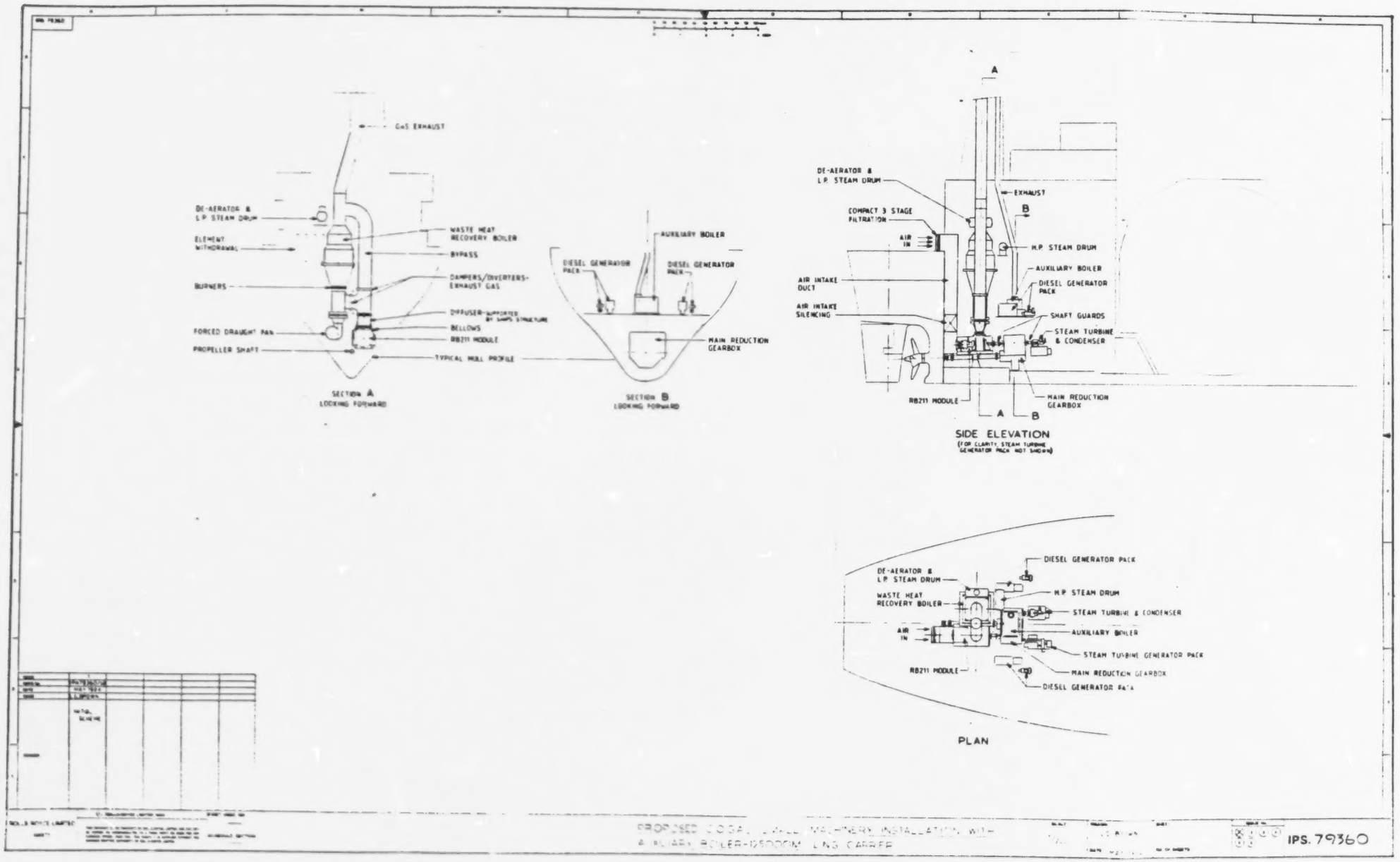


Figure 22 SHIP'S MACHINERY INSTALLATION - COGAS (SMALL)

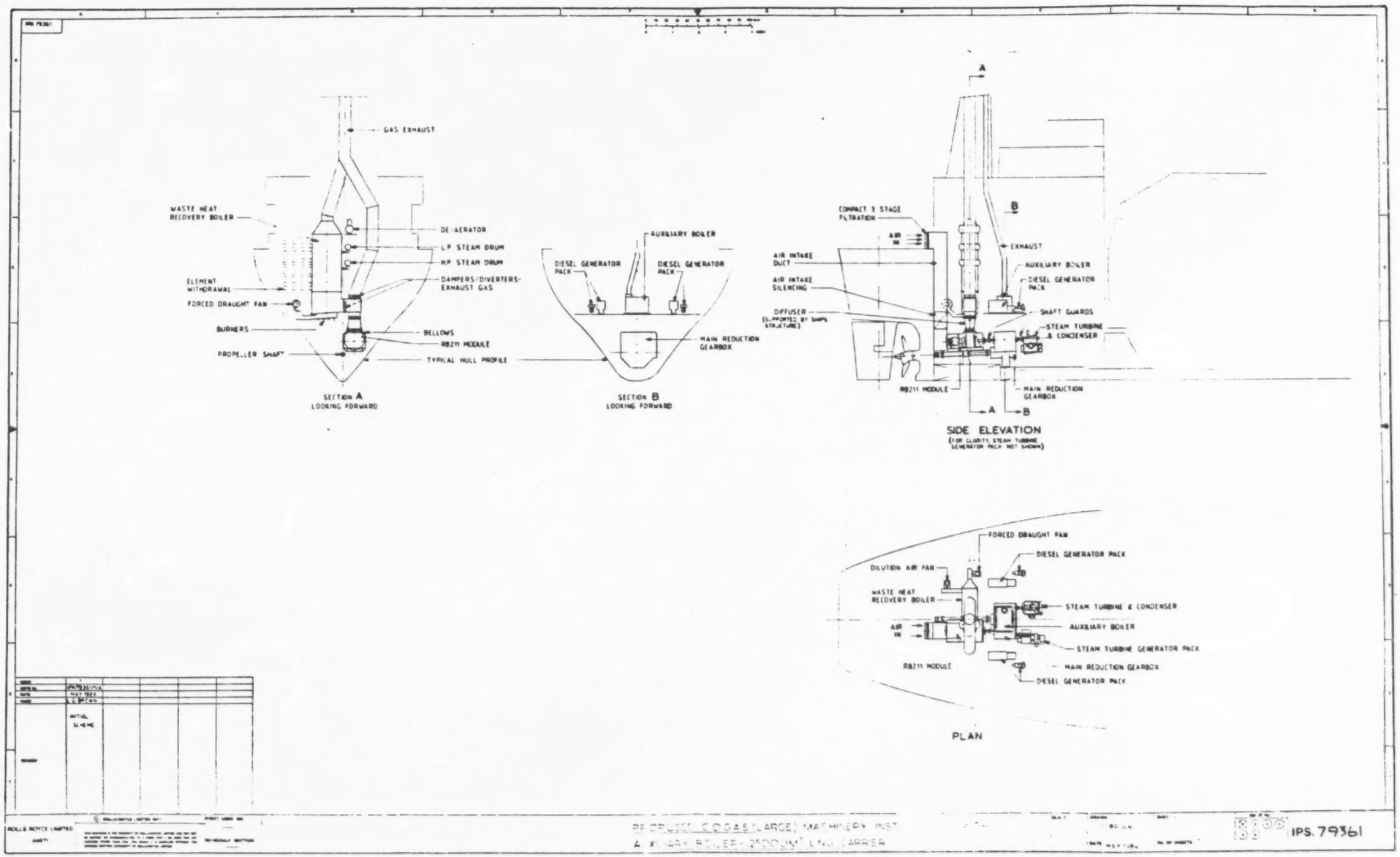


Figure 23 SHIP'S MACHINERY INSTALLATION - COGAS (LARGE)

directed to bypass the WHRB and go up the exhaust stack.

5.2.2 Air Intake System

a) The filter housing is mounted on the main deck as shown and incorporates the following:

- Access panel on the roof to facilitate gas generator removal.

- A compact 3 stage air inlet filtration system which is sized for a through velocity of 7 to 9 m/s.

- Emergency bypass doors (if filters block up) if required - not shown on drawing.

b) The air intake silencing section is designed for a through velocity of 30m/s and the length is subject to noise level required by customer.

c) The air leaves the silencing section and enters the plenum chamber thence to the gas generator intake flare.

d) Shipbuilder's supply - Filter Housing, air intake silencing, plenum chamber and trunking.

5.2.3 Exhaust System

A diffuser is fitted to the module exhaust volute bellows to change the gas path from rectangular to circular, designed to give as smooth a gas flow as possible to reduce losses. The diffuser is to be supported by the ship's structure allowing for movements and thermal expansion. The weight of the diffuser is to be fully taken by the ship's structure.

The exhaust is unsilenced, in line with latest Rolls-Royce Limited practice, and for protection the entire length is lagged to a thickness of approximately 100 mm.

5.2.4 WHRB

The RB.211/RT56 module exhaust gases are diverted through the WHRB heating the water in the tubes to produce steam for:

a) propulsion steam turbine

b) ship's steam services

c) steam turbine generator for ship's electrical power whilst in harbour

d) steam turbine driven compressor (supplying pressurised LNG to RB.211 gas generator)

The WHRB has the following features:

- HP steam drum

- LP steam drum

- deaerator

- dampers/diverters - to divert RB.211/RT56 module exhaust gases through the boiler to produce steam or if the boiler is inoperative, diverts the exhaust gases straight up the exhaust stack, bypassing WHRB.

- burners at base of boiler - used to fire the WHRB when:

  i) gas generator/power turbine are shut down

  ii) in harbour to burn off excess LNG (steam produced used for ship's steam services and steam turbine generator for ship's electrical power requirements)

- fans:

  i) dilution fan to burners for air/fuel mixture ratio.

  ii) forced draught fan, if natural circulation system is not used.

5.2.5 Auxiliary Boiler

Used in emergency conditions to maintain ship's steam services.

When running WHRB during ship's cruising, there could be a shortfall in the output of LP steam for services which can be made up by the auxiliary boiler.

5.2.6 Steam Turbine

Receives superheated steam from the WHRB and augments the output of the gas turbine by rotating the propeller shaft via the main reduction gearbox.

5.2.7 Steam Turbine/Generator Pack

Provides ship's electrical power requirements:

a) When at sea - generator driven by a main reduction gearbox.

b) When in harbour - WHRB provides steam to auxiliary steam turbine to drive generator (clutch in line to isolate system from main reduction gearbox).

c) For LNG compression when gas generator is running.

5.2.8 Diesel Generator

Two diesel generators and synchronous compensators are required to produce electrical power for bow thruster and for loading and unloading when in harbour.

5.2.9 Main Reduction Gearbox

This is not part of the submission but is drawn in to show how system works as a whole and for space requirements.

5.2.10 Additional Information

a) The hull profile shown is typical, being based upon other 125 000 m^3 LNG carriers in the absence of a specific hull profile.

b) The drawing shows a suitable COGAS installation arrangement within the confines of the hull profile in the absence of a specific hull profile.

c) The COGAS LNG Carrier is propelled by a CPP.

d) Weights (dry)

- RB.211/RT56 module	41 288 kg
- WHRB	
- Auxiliary boiler	9679 kg
- Propulsion steam turbine	
- Steam turbine/generator pack	21 772 kg
- Diesel generator pack	29 900 kg
- Main reduction gearbox	137 160 kg

e) For schematic diagram see Fig. 24.

5.3 Description of 'All-Steam' Installation Drawing (Fig. 25)

The installation consists of the following:-

- Reheat boiler
- Auxiliary boiler and superheater
- Propulsion steam turbine
- Steam turbine/generator pack
- Diesel generator pack
- Main reduction gearbox (not part of submission/study)
- Additional information

5.3.1 Reheat Boiler

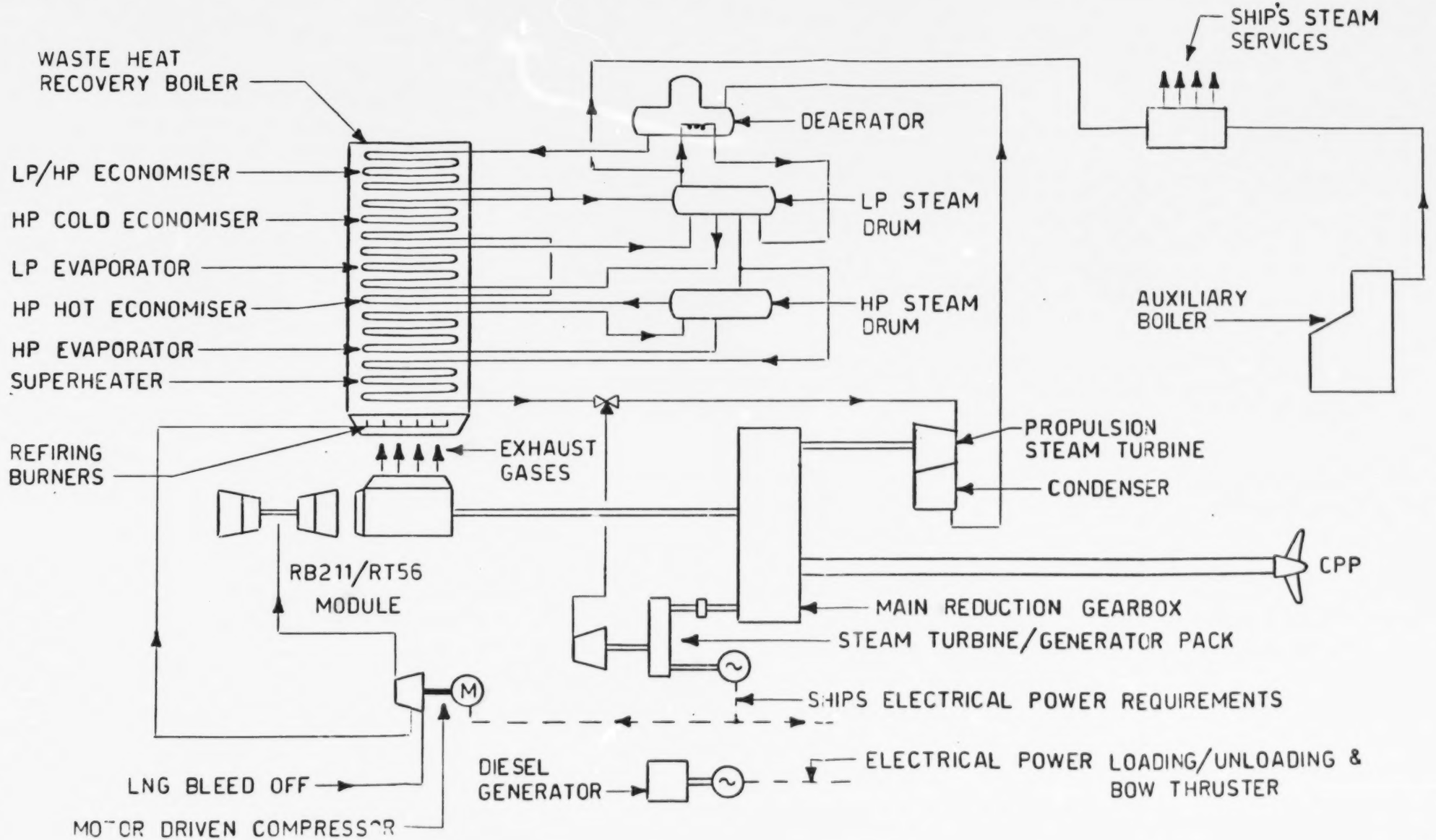


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Figure 24 SCHEMATIC - COGAS PLANT

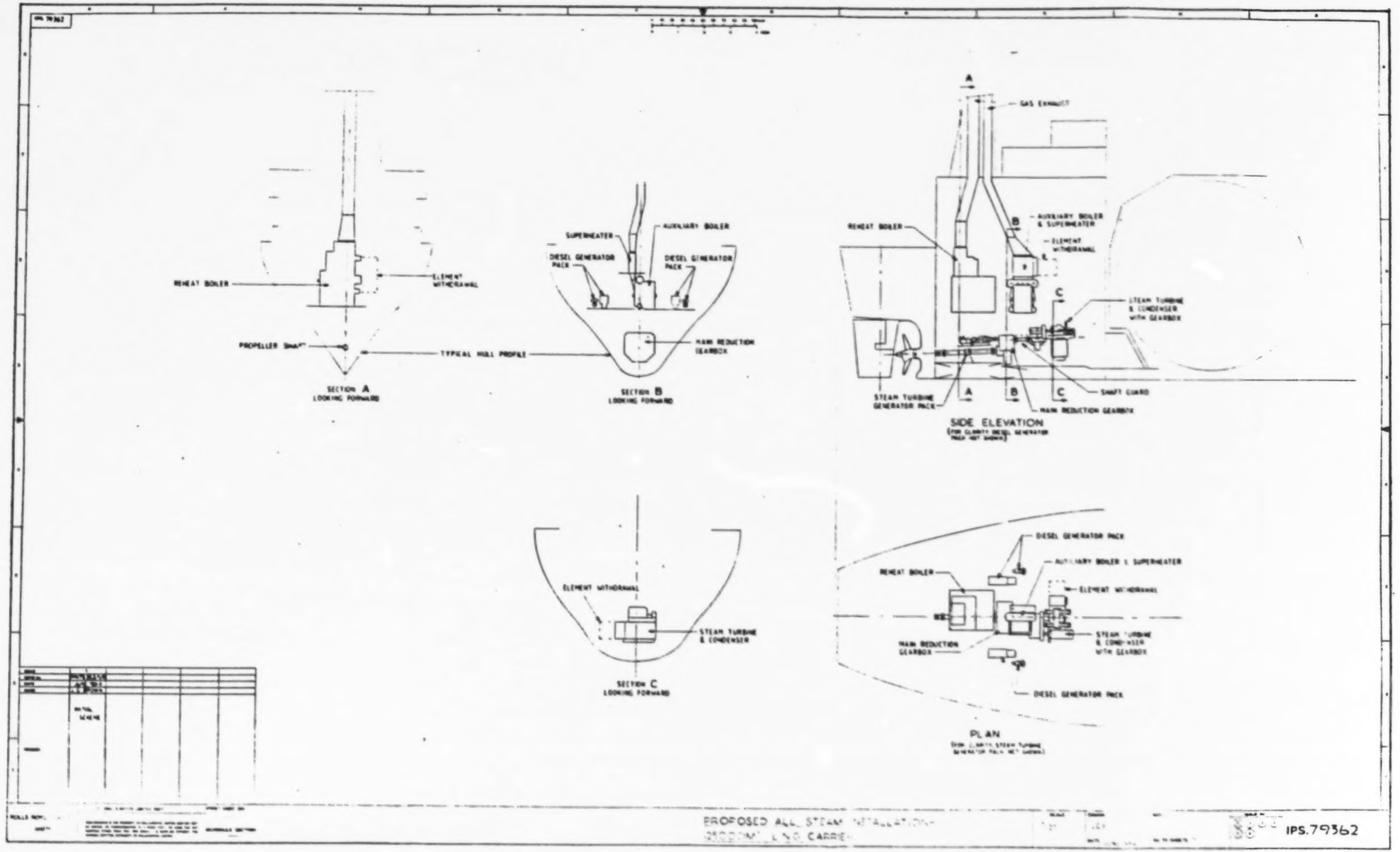

Figure 25 SHIP'S MACHINERY INSTALLATION - 'ALL-STEAM'

The forced draught, roof fired boiler (burning gas or oil fuel) heats the water in the tubes to produce steam for (a) propulsion steam turbine (HP, IP and LP) and (b) ship's steam services.

The reheat boiler has the following features:-

- Reheater with reheat control dampers
- Dampers - reheat steam control is affected by dampers situated above reheater and bypass economiser
- Base tube and extended surface economisers
- Rotary gas airheater
- Burners
- Sootblowers
- Forced draught fan - air flow through boiler also for protection of reheater when need for reheater, during extended periods of steaming, is not required (such as astern operation and extended cargo pumping).

5.3.2 Auxiliary Boiler and Superheater

When in harbour

Burners fired by excess LNG or diesel oil to provide steam for (a) ship's steam services, (b) steam turbine generator for ship's electrical power requirements.

When at sea

If reheat boiler is inoperative then the auxiliary boiler can be operated to provide steam for (a) propulsion steam turbine (HP stages), (b) ship's steam services.

Superheater

The superheater is situated directly above the auxiliary boiler, connected by a suitable ducting, and mounted to the ship's structure.

5.3.3. Propulsion Steam Turbine

The steam turbine receives superheated steam from the reheat boiler or in an emergency condition, from the auxiliary boiler. The steam turbine consists of HP and IP turbines coupled together on a common shaft and a separate LP turbine. The two output shafts go into a reduction gearbox to give a single output speed to drive into the main reduction gearbox. The astern turbine is incorporated into the LP turbine.

5.3.4 Steam Turbine/Generator Pack

Provides ship's electrical power requirements

a) When at sea - generator driven via main reduction gearbox

b) When in harbour - auxiliary boiler provides steam to auxiliary steam turbine to drive generator - clutch in line to isolate system from main reduction gearbox.

5.3.5 Diesel Generator Pack

Two diesel generators and synchronous compensators are required to produce electrical power for bow thruster and for loading and unloading when in harbour.

5.3.6 Main Reduction Gearbox

This is not part of the submission but is drawn in to show how the system works as a whole and for space requirements.

5.3.7 Additional Information

a) The hull profile shown is typical, being based upon other 125 000 m^3 LNG carriers in the absence of a specific hull profile.

b) The drawing shows a suitable 'all-steam' installation arrangement within the confines of the hull profile.

c) The 'all-steam' LNG carrier is propelled by a FPP.

d) Weights (dry)

- Reheat boiler	
- Auxiliary boiler and superheater	
- Propulsion steam turbine	
- Steam turbine generator pack	21 772 kg
- Diesel generator pack	29 900 kg
- Main reduction gearbox	132 090 kg

e) For schematic diagram see Fig. 26.

5.4 Ship Performance

The ship's operational requirements as listed in Section 2 of this document are unchanged and provide the basis by which the annual operation of the ship using the various machinery options can be judged.

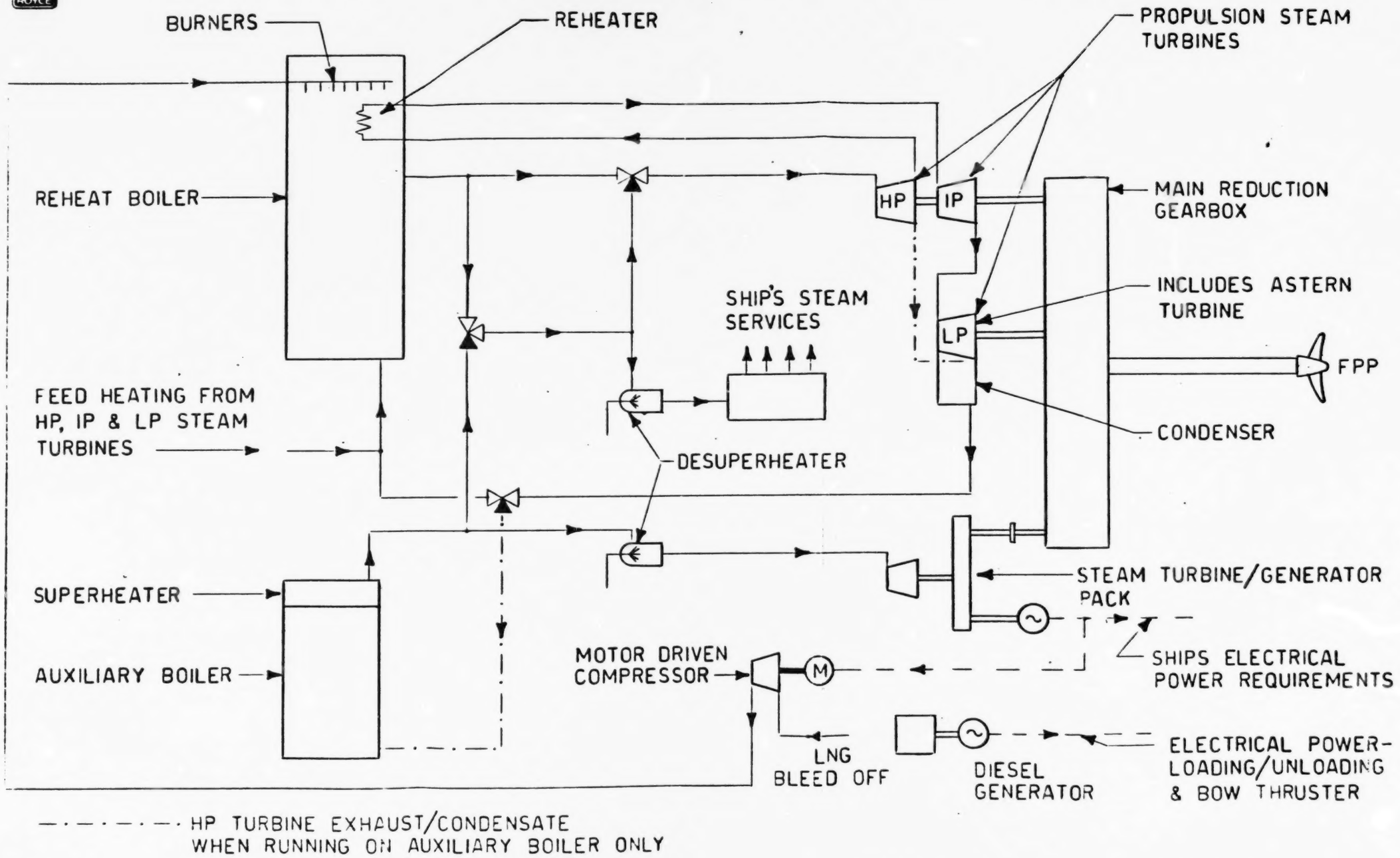

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Figure 26 SCHEMATIC - 'ALL-STEAM' PLANT

The voyage profile (Section 2.2.6) sets out the journey times and ship's speeds for the outward (loaded) departure from Canada, a distance of 4310 nautical miles.

The return (ballast) trip home from Japan is a distance of approximately 4288 nautical miles.

The air temperature, as already discussed, has been fixed at 9 deg.C. as an average working ambient for design and performance calculation. The seawater temperature settled on as a working value after studying variations across the sea route (Figs. 27 and 28) was also 9 deg.C.

Method Analysis

Waste heat recovery boiler evaporation rates at specific gas turbine ratings were matched to the steam turbine steam requirements at 9 deg.C. This enabled the combined shaft power of the plant at the gearbox input to be realised. Services' steam power available from the waste heat boiler at 800 kPa, 4 T/h, 170 deg.C. was assumed to return to the main steam turbine condenser.

Overall thermal efficiency (η_{OT}) is defined as:

i) $$\eta_{OT} = \frac{GTHP+STHP+SER}{FUEL}$$

ii) Thermal efficiency - propulsive (η_{PT}) is defined as:

$$\eta_{PT} = \frac{PSP}{FUEL}$$

Where:

GTHP = gas turbine power at gearbox input
STHP = steam turbine power at gearbox input
SER = services power supplied from boiler
FUEL = fuel energy entering gas turbine
PSP = shaft power available for propulsion

Using the waste heat recovery boiler and steam turbine data in this manner performance curves of specific fuel consumption versus ship speed were produced for each type of plant (Figs. 29 to 32).

Working from the above mentioned curves and applying these to the ship voyage profile an appreciation of the overall LNG consumption could be gained.

Fig. 33 lists the efficiency of the plants at various stages through the machinery package and also gives a value for the predicted level of LNG make up required, for the round trip of 25 days. These figures are based on the ship design condition with an air and sea temperature of 9 deg.C. respectively.

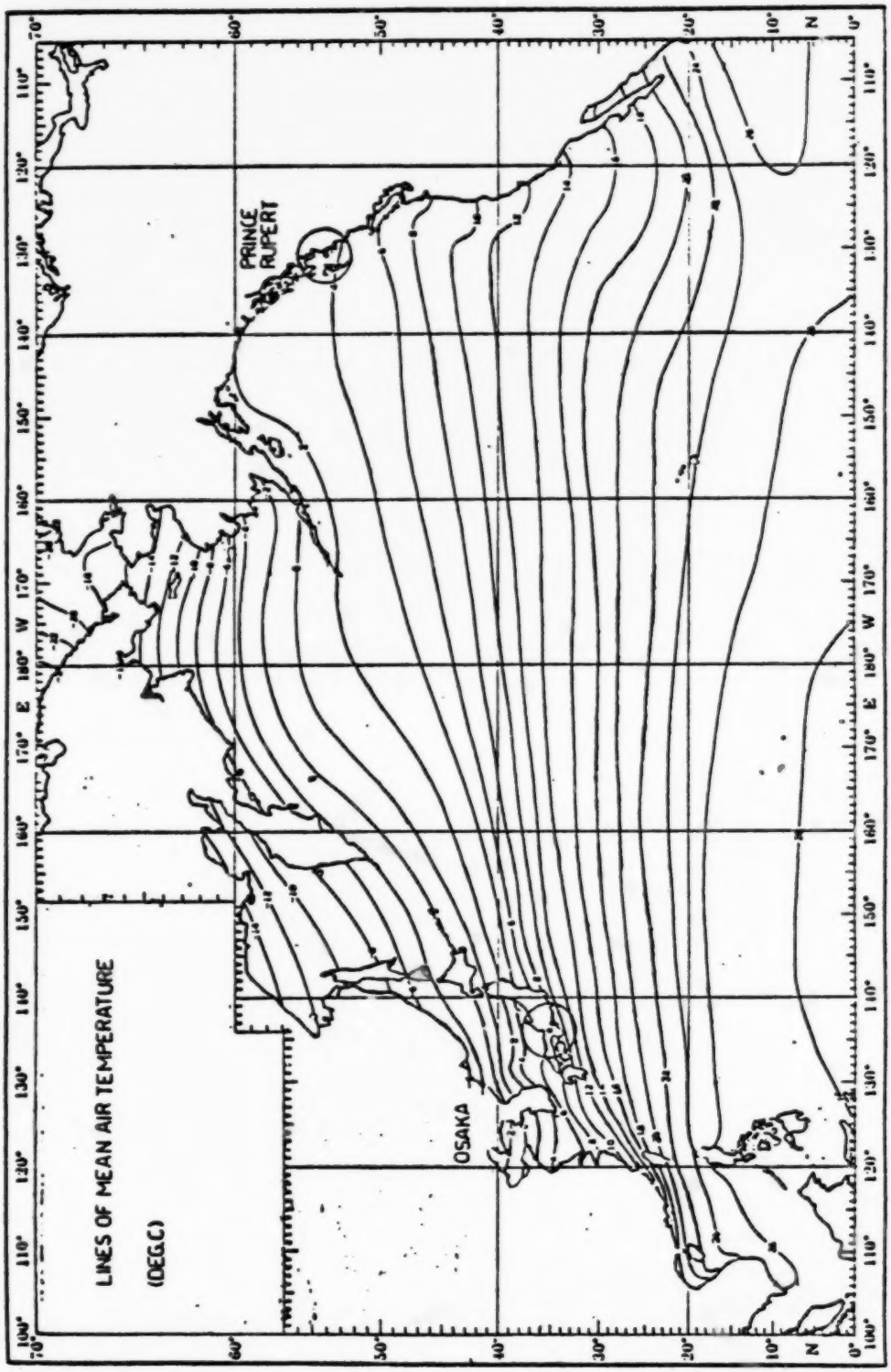


Figure 27 JANUARY SURFACE AIR TEMPERATURE

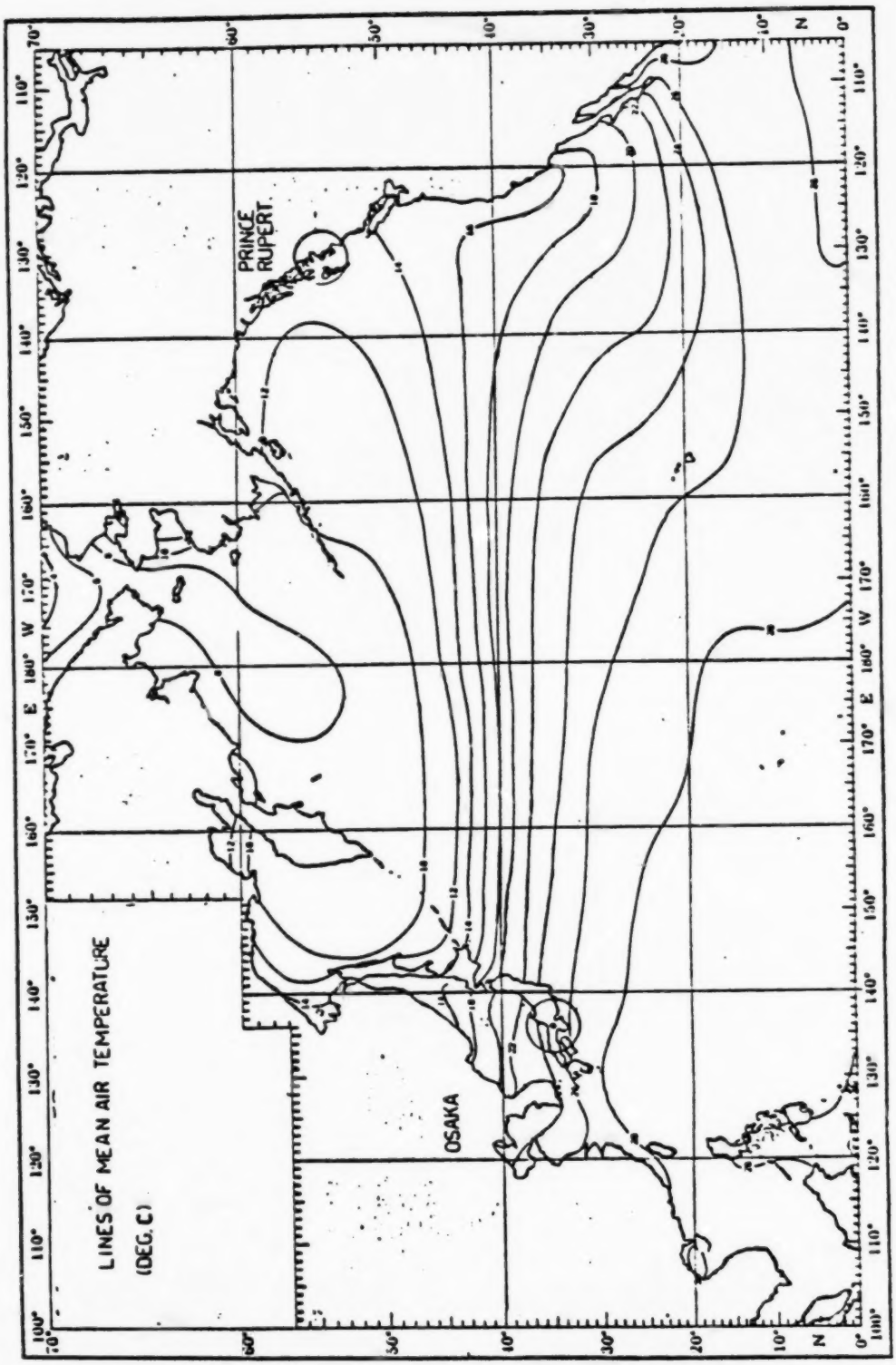
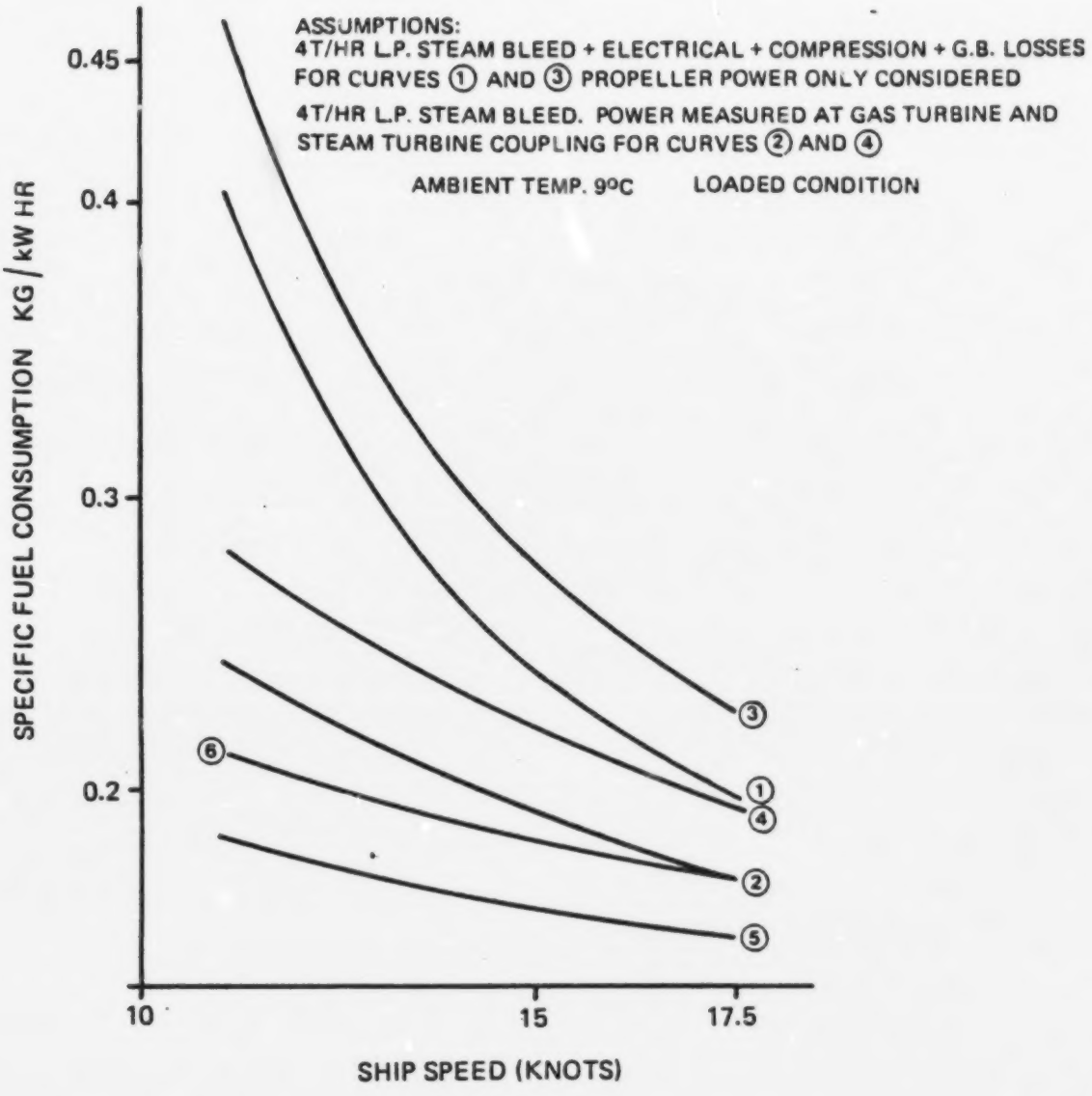


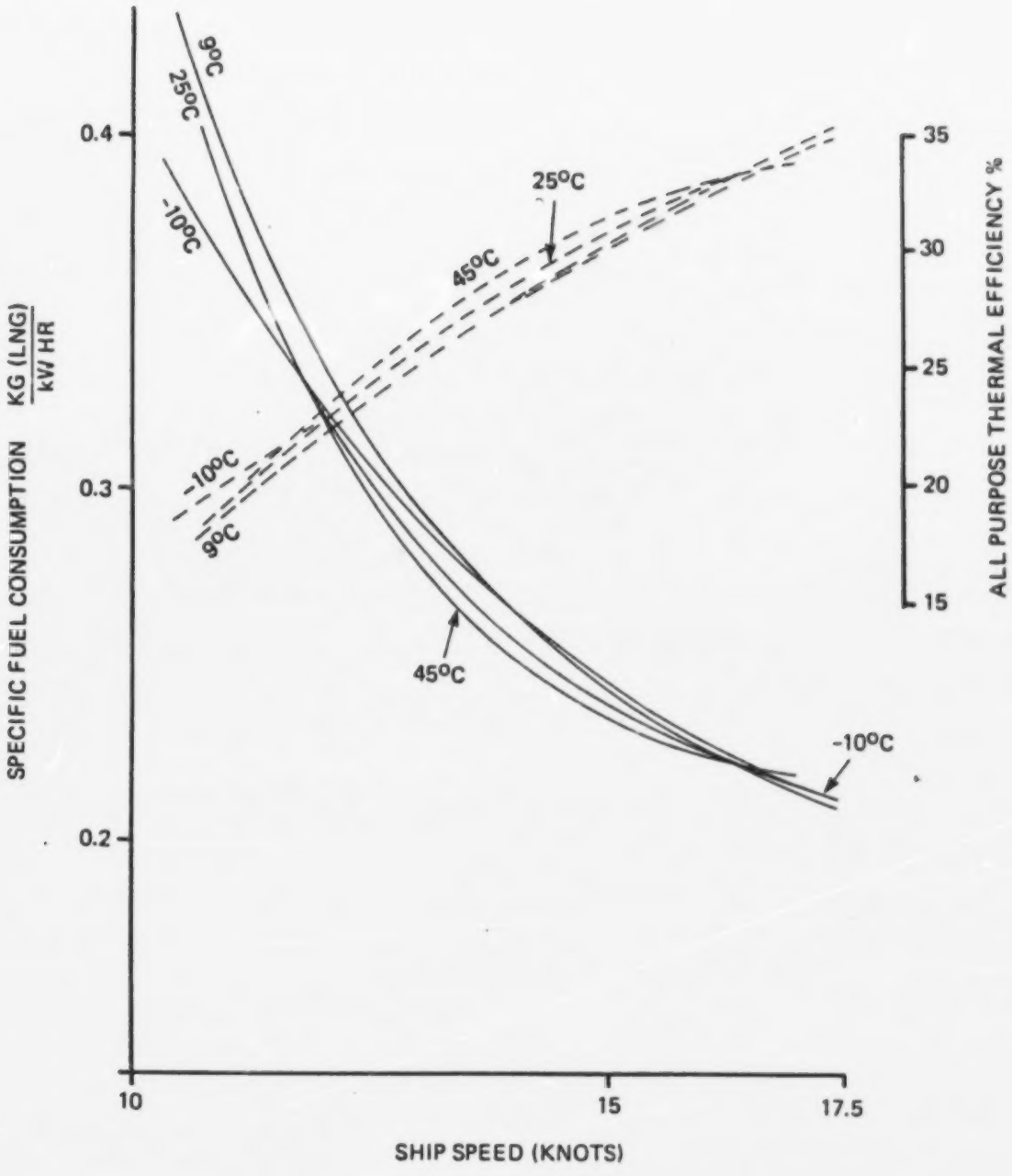
Figure 28 JULY SURFACE AIR TEMPERATURE



VARIATION OF SPECIFIC FUEL CONSUMPTION WITH SHIP SPEED
9°C AIR INTAKE, 9°C SEAWATER TEMP.

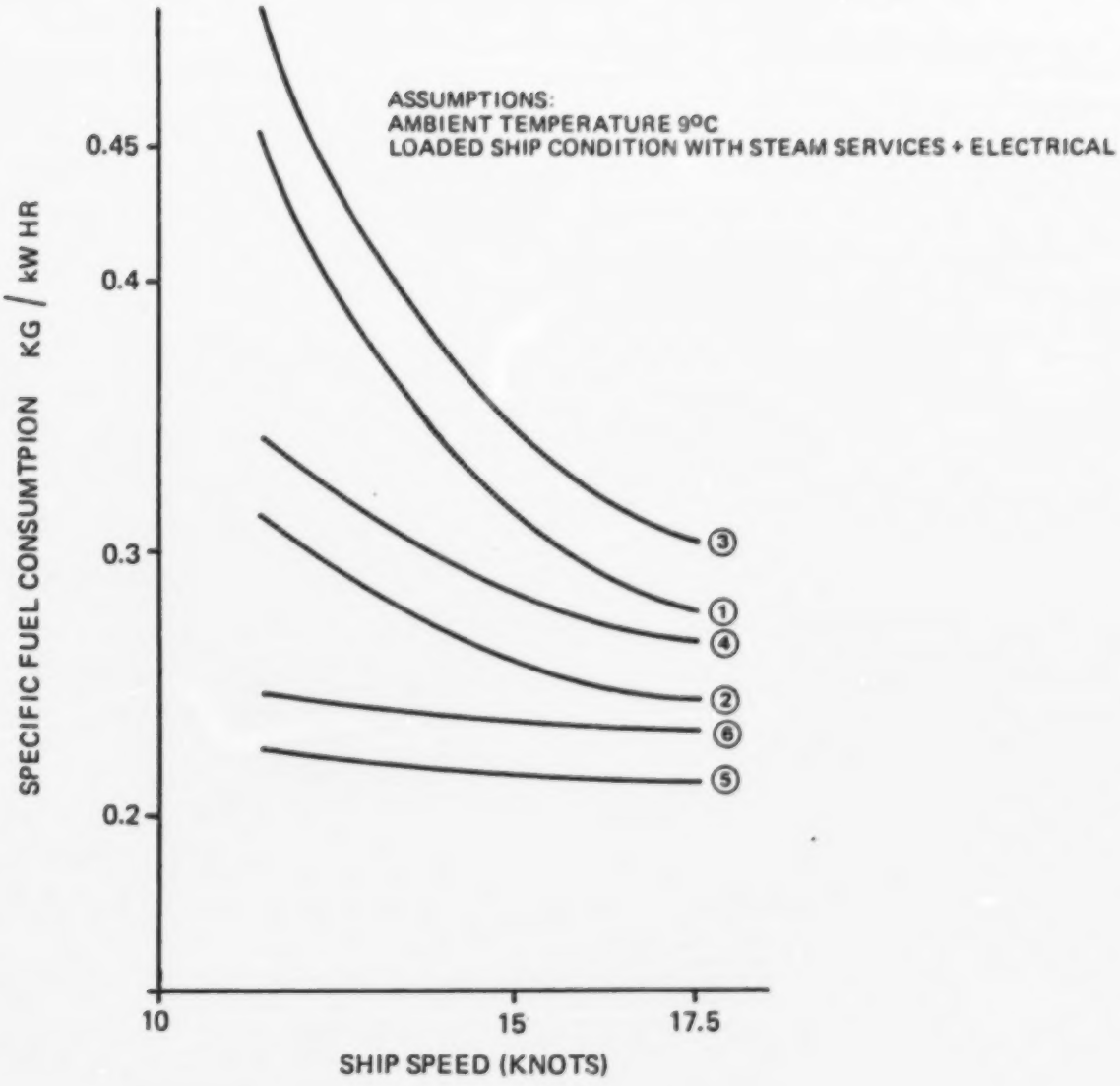
① S.F.C. ALL PURPOSE REFERRED TO PROPELLER (LNG)
② S.F.C. REFERRED TO GEARBOX INPUT (LNG)
③ S.F.C. ALL PURPOSE REFERRED TO PROPELLER (DIESEL)
④ S.F.C. REFERRED TO GEARBOX INPUT (DIESEL)
⑤ S.F.C. OVERALL THERMAL (LNG)
⑥ S.F.C. OVERALL THERMAL (DIESEL)

Figure 29 LARGE COGAS
ECONOMIC LNG CARRIER



S.F.C. AND EFFICIENCY (ALL PURPOSE REFERRED TO PROPELLER) V SHIP SPEED (KNOTS) FOR -10°, 9°, 25° AND 45° CENTIGRADE AIR INTAKE TEMP.

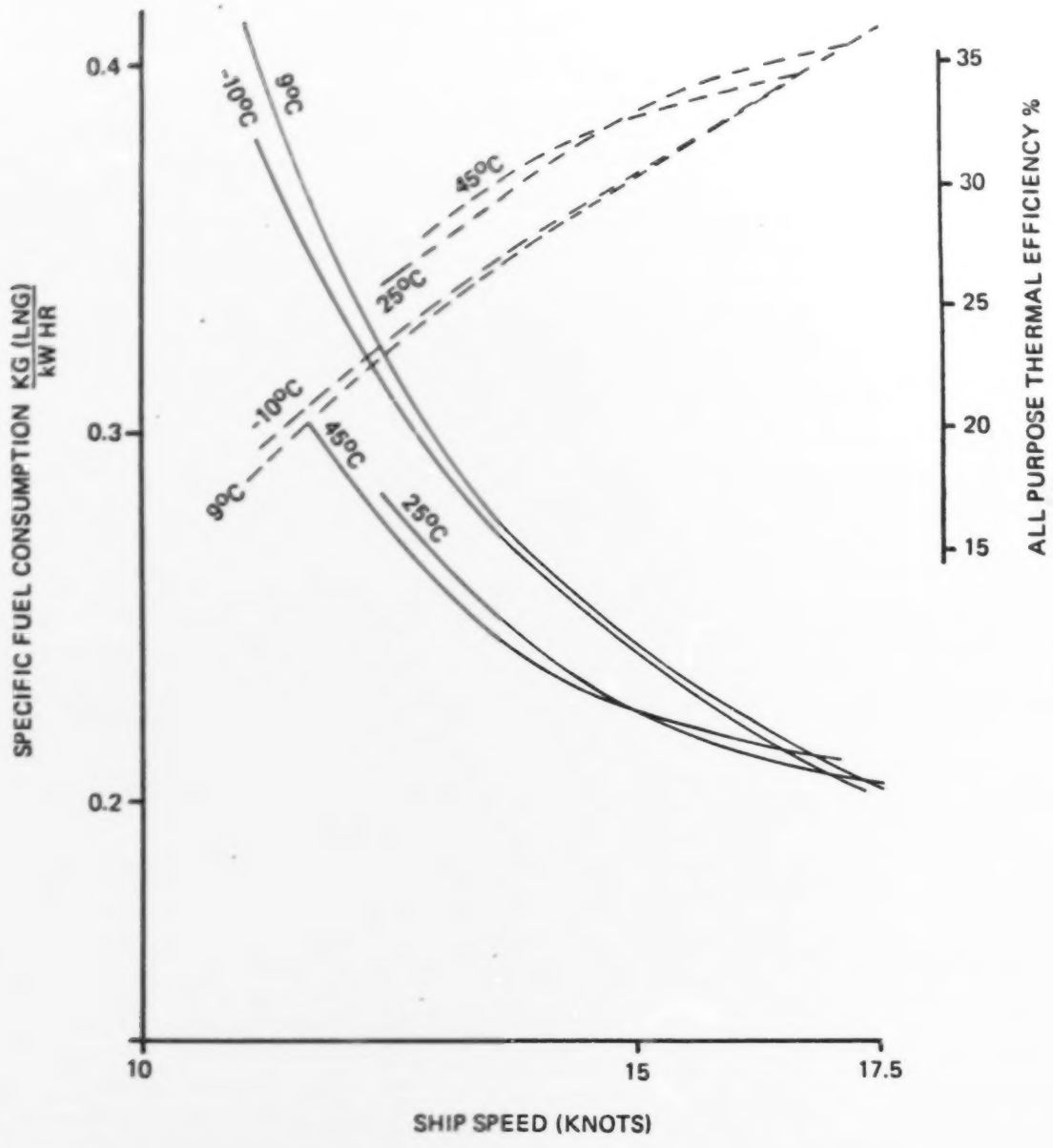
Figure 30 SMALL COGAS PLANT
ECONOMIC LNG CARRIER



VARIATION OF SPECIFIC FUEL CONSUMPTION WITH SHIP SPEED
9°C AIR INTAKE, 9°C SEAWATER TEMP.

① S.F.C. ALL PURPOSE REFERRED TO PROPELLER (LNG)
② S.F.C. REFERRED TO GEARBOX INPUT (LNG)
③ S.F.C. ALL PURPOSE REFERRED TO PROPELLER (DIESEL)
④ S.F.C. REFERRED TO GEARBOX INPUT (DIESEL)
⑤ S.F.C. OVERALL THERMAL (LNG)
⑥ S.F.C. OVERALL THERMAL (DIESEL)

Figure 31 ALL STEAM REHEAT
ECONOMIC LNG CARRIER



S.F.C. AND EFFICIENCY (ALL PURPOSE REFERRED TO PROPELLER) V SHIP SPEED (KNOTS) FOR -10°, 9°, 25° AND 45° CENTIGRADE AIR INTAKE TEMP.

Figure 32 LARGE COGAS PLANT
ECONOMIC LNG CARRIER

Figure 33 SHIP PERFORMANCE AT 9°C

	SMALL COGAS NEI W.H.R.B. WESTINGHOUSE STEAM TURBINE 16 816 kW	LARGE COGAS 16 331 kW				ALL STEAM PLANT GEC REHEAT STEAM TURBINE FOSTER WHEELER REHEAT BOILER
		FW/ WHSE	FW/ GEC	GREENS WHSE	SUNROD /WHSE	
Overall Thermal Efficiency at gearbox input (%)	47	49	49	49	49	34,7
Thermal Efficiency at gearbox input shaft power only (%)	41,6	43	43	43	43	30,3
All Purpose Thermal Eff. at propeller (%)	35,2	36,5	36,5	36,5	36,5	26,5
LNG make-up per round trip (Tonnes)	307	262	267	258	259	910
Diesel make-up per round trip (Tonnes)	353	300	206	296	297	999

Performance Comment

For the 'large' COGAS plant LNG consumption exceeds the boil-off by approximately 8,9% in the loaded mode. In the case of the 'all-steam' reheat plant consumption exceeds boil-off by 50,9% in the same ship condition. In the ballast condition where the boil off rate is reduced from 0,015% to 0,012% per day the amount by which LNG consumption exceeds the boil off is correspondingly more than in the loaded condition for all types of plant.

From Fig. 33 it can be seen that the 'large' COGAS design will require the minimum additional fuel make-up, slightly less than the 'small' COGAS design. Both COGAS plants need considerably less make-up fuel than the 'all-steam' option - less than one third of the annual running requirement.

Where LNG consumption exceeds the boil off flow from the storage tanks extra LNG can be generated by forced vaporizing some of the cargo. An alternative to this would be to utilise the dual fuel capability of the RB.211/RT56 gas turbine and supplement with marine distillate fuel.

Typical heat balance diagrams (Figs. 34 and 35) show pressure, temperature, mass flows and enthalpy at a design point of 9 deg.C., for both COGAS and 'all-steam' plants.

5.5 Overhaul, Maintenance and Lifing Criteria

5.5.1 RB.211/RT56 Module

Major parameters which influence lifing of the module are:

a) Starting frequency (ie., hours run per start).

b) Impurities (eg., Sulphur, Sodium, Vanadium, etc.) in fuel and air supply.

c) Rating at which engine is run.

d) Materials used in the construction of the gas turbine.

As a general rule, it can be said that the greater the value of any of the first three parameters listed above, the shorter the overhaul life becomes.

In common with all types of gas generators, the RB.211 requires periodic inspection, maintenance, and reconditioning to maintain maximum performance, reliability and availability of the unit. It has no fixed overhaul life since it has been designed from the outset to operate on an 'on-condition' basis, which means it can continue in operation until, by systematic health or condition monitoring of operating parameters, a deterioration factor is indicated.

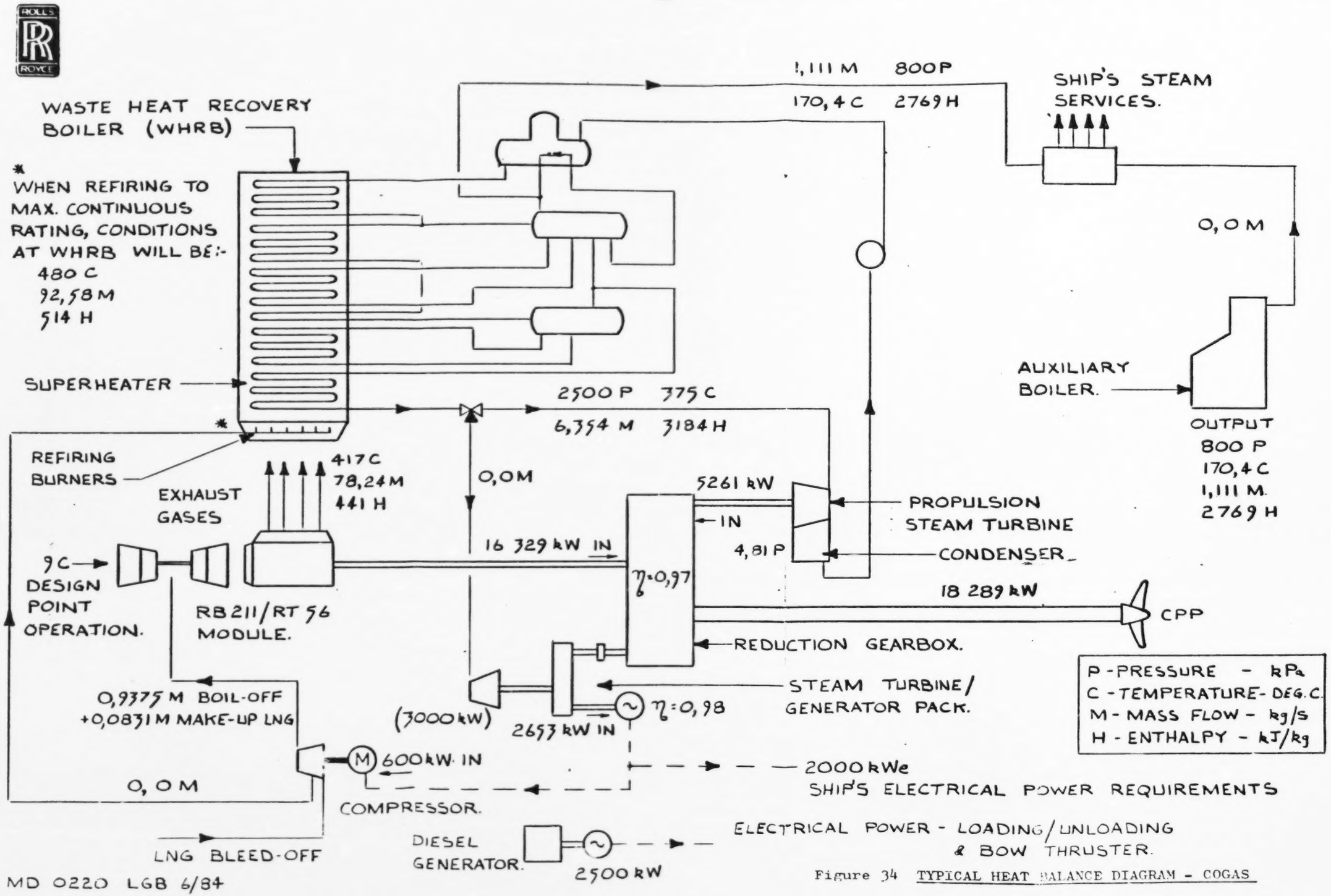


Figure 34 TYPICAL HEAT BALANCE DIAGRAM - COGAS

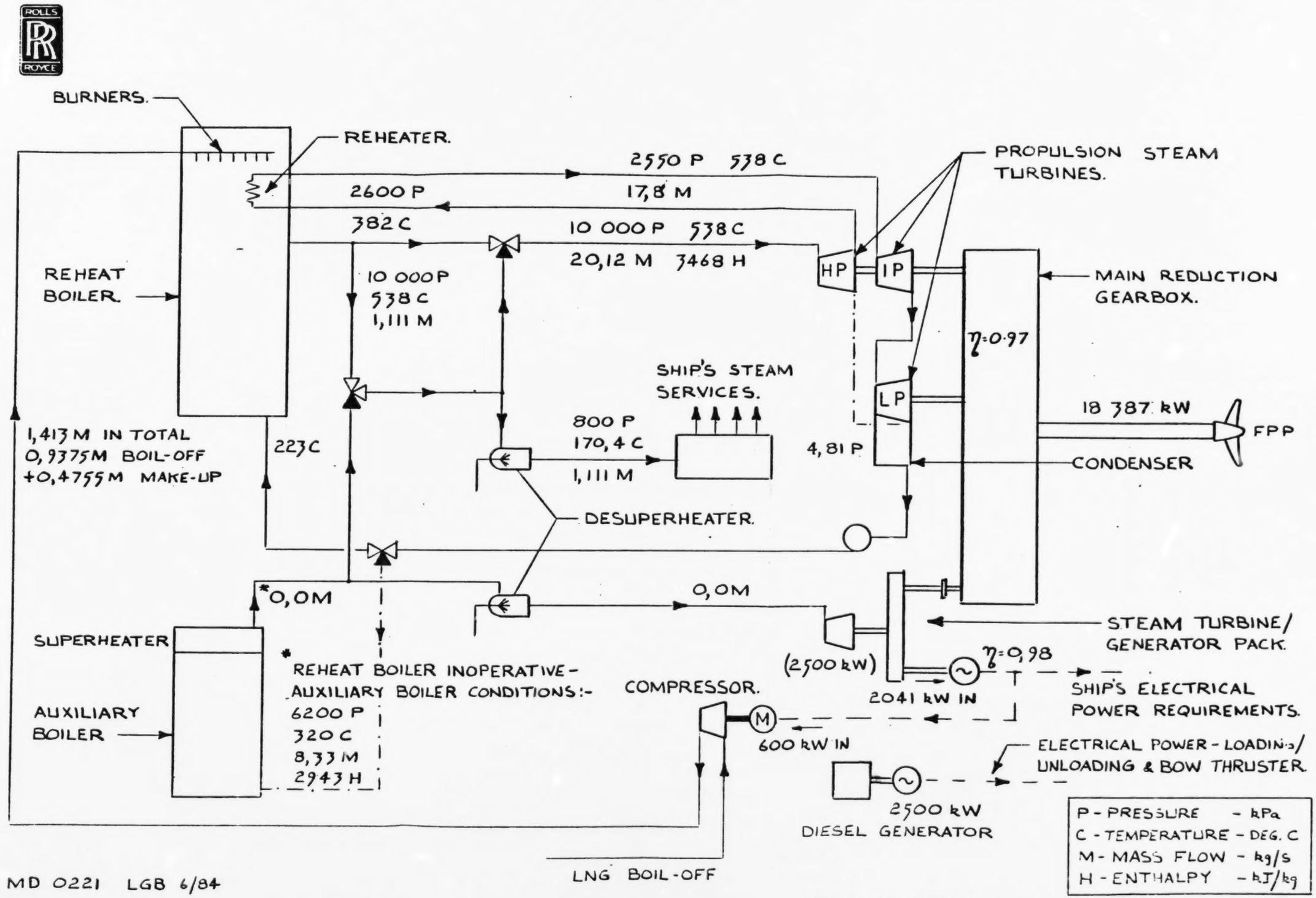


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Figure 35 TYPICAL HEAT BALANCE DIAGRAM - 'ALL-STEAM'

The three main categories of parameter to be condition monitored are:

Mechanical - vibration levels, oil consumption and acidity, hydraulic and lubricating oil condition, and metal wear

Performance - pressures and temperatures

Physical Condition - visual and boroscopic inspections

By recording and analysing the trend and changes in the values of these gas generator running parameters, evidence of deterioration can be identified sufficiently early to allow for planned maintenance. This procedure enables a maximum economic installed life to be achieved.

The RT56 power turbine has a long life capability of between 27 000 hours to 100 000 hours, depending on the ship's operating profile and propulsion transmission system.

The RT56 power turbine is designed with the same philosophy of engine health monitoring procedures and capability incorporated as with the RB.211 gas generator.

5.5.2 Boilers

Major parameters affecting lifing of boilers are:

a) Impurities in fuel, air and steam.

b) Materials used in construction of boilers.

c) Excess air.

d) Boiler tubing wall thickness.

High temperature oil-ash corrosion, especially with regard to the boiler tubes, is one feature which needs to be carefully examined when considering lifing of boilers. The corrosion rate increases with temperature, Vanadium content in fuel and excess air level.

Typical boiler maintenance problems are:

- Corrosion problems with the boiler (try to keep heating surfaces clean).
- Steam piping repairs.
- Malfunctions of combustion control.
- Failures of warm-up steam line valves.

Boiler maintenance is usually limited to routine inspections and replacement as necessary of normal wearing items such as pump bearings. During inspections it is important that records are kept of pressures and temperatures so that any unusual changes in these parameters become apparent immediately, and thus provide prior warning of impending maintenance.

5.5.3 Steam Turbines

Critical parameters which influence the lifing of steam turbines are:

a) Turbine steam purity.

b) Suspended solids count in steam.

c) Materials used in construction of steam turbine.

Corrosion fatigue, corrosion and stress corrosion are the undesirable characteristics associated with impurities in the steam. Additionally, disposition of these impurities is likely to have an unfavourable effect by clogging seals and clearances in valves, upsetting pressure distributions, and lowering the efficiency of the blades. Thus, it is essential to monitor and analyse, on a continuous basis, the purity of the steam supply, and when necessary to take the appropriate action to ensure that the steam purity is maintained.

Solids in the steam supplied to the turbine can, when in excess of one part per million cause turbine erosion.

6. BUDGETARY COSTS (Study Phase 5)

This Section deals with the following items:

- Plant Acquisition costs (Fig. 36)
- Maintenance requirements and cost
- Cost of make-up fuel and maintenance costs for a ten year operation period
- Manning requirements
- Canadian manufacturing content (Fig. 37)
- Fuel Consumption and cost for loaded and ballast legs of voyage (Fig. 38)
- Annual fuel costs and consumption per ship (Fig. 39)

6.1 Plant Acquisition Costs

Acquisition budgetary costs were obtained from suppliers and these are shown tabulated in Fig. 36 for the COGAS and the 'all-steam' plants. The main options A, B, and C correspond to the main options as selected in the Ship Performance and Operational Assessment, Section 5.

The cost of the propellers and shafting is included to reflect the difference between a controllable pitch propeller required on the COGAS system to satisfy the reversing operation with a gas turbine and the fixed pitch propeller on an 'all-steam' plant where a reversing facility is included in the steam turbine arrangement.

No costs are shown for a main gearbox or control system because it is considered that the costs for such items would be similar.

The shipping and insurance costs were obtained from Rolls-Royce shipping agents and consider the cost of shipping of a complete propulsion plant package.

Although the individual manufactures components vary in cost, the tables show the overall plant cost for both the COGAS and 'All-steam' systems is in the order of 10 million Canadian dollars. For this reason therefore, it was considered unnecessary to undertake an amortisation analysis.

PLANT ACQUISITION COSTS

(Canadian dollars (Million) per ship set)

COGAS

	OPTION A				OPTION B				
	Supplier	COSTS Main Plant	COSTS Spares, Tooling	Shpg: Ins.	Supplier	COSTS Main Plant	COSTS Spares, Tooling	Shpg: Ins.	
Main Boiler	N.E.I. WH RB	1,67	0,03		Foster Wheeler WH RB	1,43	0,045		NOTES: The budgetary costs in Canadian dollars are based on an exchange rate of 1,8 C $ to 1,0 pound sterling. For "Sunrod" an exchange rate of 6,5 Swedish Krona to 1,0 Canadian dollars has been assumed.
Auxiliary Boiler	N.E.I.				Greens	0,07	0,002		
Main Steam Turbine	Westinghouse	0,55	Incl. in main cost	0,16	Westinghouse	0,55		0,15	The cost of the main plant is generally given FOB at the port of the country of manufacture.
Auxiliary Steam Turbine Condenser	Westinghouse	0,63	Incl. in main cost		Westinghouse	0,63	Incl. in main cost		Shipping cost estimates allow for shipping from the country of manufacture to an assumed shipbuilder in Kobe, Japan, or similar. An allowance for crating, where applicable, has been made in the shipping costs.
Generator	Westinghouse				Westinghouse				Options A and B are considered to be the main options for the COGAS plant, the remaining three are shown for comparison.
Propellers and Shafting	Stone Vickers	0,86	Incl.in main cost		Stone Vickers	0,86	Incl. in main cost		Spare provisioning for the RB 211/RT 56 considers a basic 5 ship fleet with base spares of one complete gas generator and two 04 module hot sections and combustion units.
Gas Turbine	RB 211 Gas Generator RT 56 Power Turbine	5,04	0,99	0,07	RB 211 Gas Generator RT 56 Power Turbine	5,04	0,99	0,07	
Sub total		8,75	1,02	0,23		8,58	1,04	0,22	
Total			10				9,84		

Figure 36 PLANT ACQUISITION COSTS (Sheet 1)

PLANT ACQUISITION COSTS
(Canadian dollars (Million) per ship set)

COGAS

	Supplier	COSTS Main Plant	COSTS Spares, Tooling	Shpg: Ins.	Supplier	COSTS Main Plant	COSTS Spares, Tooling	Shpg: Ins.	Supplier	COSTS Main Plant	COSTS Spares, Tooling	Shpg: Ins.
Main Boiler	Foster Wheeler	1,43	0,045		Greens	0,87	0,03		Sunrod	2,86	0,1	
Auxiliary Boiler	Greens	0,072	0,002		Greens	0,072	0,002		Sunrod	0,05	0,0018	
Main Steam Turbine	G.E.C.	0,814	Incl.in main cost	0,18	Westinghouse	0,55	Incl.in main cost	0,12	Westinghouse	0,55	Incl.in main cost	0,15
Auxiliary Steam Turbine and Condenser Generator	Westinghouse Westinghouse	0,63	Incl.in main cost		Westinghouse Westinghouse	0,63	Incl.in main cost		Westinghouse Westinghouse	0,63	Incl.in main cost	
Propellers & Shafting	Stone Vickers	0,86	Incl.in main cost	-	Stone Vickers	0,86	Incl.in main cost		Stone Vickers	0,86	Incl.in main cost	
Gas Turbine	RB 211 RT 56 Power Turbine	5,04	0,99	0,07	RB 211 Gas Generator RT 56 Power Turbine	5,04	0,99	0,07	RB 211 Gas Generator RT 56 Power Turbine	5,04	0,99	0,07
Sub total		8,85	1,04	0,25		8,02	1,02	0,19		9,99	1,092	0,22
Total			10,14				9,23				11,3	

Figure 36 PLANT ACQUISITION COSTS (Sheet 2)

PLANT ACQUISITION COSTS

(Canadian dollars (Million) per ship set)

ALL-STEAM

	OPTION C								
	Supplier	COSTS			Supplier	COSTS			
		Main Plant	Spares, Tooling	Shpg: Ins.		Main Plant	Spares, Tooling	Shpg: Ins.	
Main Boiler	Foster Wheeler	2,63	0,09		Sunrod	4,62	0,16		NOTES: Option C is considered as the main contender for the 'All-Steam' plant. The other plant combination is shown for comparison
Auxiliary Boiler	Foster Wheeler	1,32	0,05	0,56	Sunrod			0,60	
Main Steam Turbine	G.E.C.	4,25	Incl. in main cost		G.E.C.	4,25	Inc. in main cost		
Auxiliary Steam Turbine and Condenser and Generator	Westinghouse	0,63	Incl. in main cost		Westinghouse	0,63	Inc. in main cost		
Propellers & Shafting	-	0,4	Incl. in main cost		-	0,4	Incl.in main cost		
Sub total		9,23	0,14	0,56		9,9	0,16	0,60	
Total		9,93				10,76			

Figure 36 PLANT ACQUISITION COSTS (Sheet 9)

6.2 Maintenance Requirements and Cost RB.211/RT56

6.2.1 RB.211 Gas Generator and RT56 Power Turbine Maintenance.

As stated in Section 5.5.1, the RB.211 is designed to operate on on-condition basis, which means that it is not mandatory to remove a component unless performance and inspection proves it to be necessary.

The RB.211 gas generator is constructed on a modular basis which enables component parts to be quickly exchanged as required. The gas generator comprises 5 major module components and 3 mini modules designated as follows:

Cold end	01, 02, 03 compressor modules
Hot end	04 and 05 turbine modules. There are also three mini modules 04/1, 04/2 and 04/3.

The inspection periods appropriate to the operation of the LNG carrier would be as follows:

Weekly	Minor running maintenance checks by ship's staff.
4000 hour intervals	Routine servicing inspection of gas generator accessories and mechanisms, change of igniter plugs and full borescope inspection of compressors, combustion and turbine areas requiring - 6 man hours.
4000 hour intervals	Routine servicing inspection of the gas generator by ship's staff - 6 man hours.
10 000 hour intervals	Short inspection of the power turbine by ship's staff - 6 man hours
16 000 hour intervals	Engine condition/inspection being a major maintenance action - 40 man hours Subject to conditions it may be necessary to change the 04 hot section module - 9 hours. Alternatively, it may be possible to change the 04 module hot section at a later 4000 hourly inspection.
24 000 hour intervals	Subject to condition assessment the gas generator may be scheduled for reconditioning at an overhaul workshop (gas generator change in 8 hours).
30 000 hour intervals	Repeat of 10 000 hours power turbine inspection together with checks on the exhaust diffuser and

approx. 4 years of ship life	second stage blades - 10 man hours.
60 000 hour intervals approx. 8 years of ship life	Major maintenance action on the RT56 power turbine rotor.

All maintenance actions may be done on board ship except for the 24 000 hour gas generator overhaul and some parts of the 60 000 hour power turbine inspection.

6.2.2 RB.211/RT56 Maintenance Costs

An overall maintenance cost of C$10,16 per hour of operation has been established by operators of Rolls-Royce RB.211/RT56 units on gas pumping offshore duties. This maintenance rate is valid for current 1983 to 1984 gas generator module removal rates.

For an LNG carrier operating 13 round trips from Canada to Japan per year this represents a cost of C$73 152 per year which is approximately 1,5 per cent per annum of the capital cost of the RB.211/RT56. Recent average costs of Rolls-Royce repairs on all of the RB.211 gas generator modules when applied to the inspection period intervals would suggest an average figure for the gas generator modules of only 0,9 per cent per annum per RB.211/RT56 module.

It is considered, therefore, that 1,5 per cent per year represents a realistic overhaul cost figure for the RB.211/RT56 module, which would adequately satisfy the maintenance requirements. Further, maintenance costs should be well contained within this figure since the power setting of the gas turbine used in the LNG carrier configuration is only 65 per cent of the maximum continuous rating, which should improve engine life.

6.2.3 COGAS Steam Plant Maintenance Cost

- Option A

Not considered because of the higher initial plant cost.

- Option B

The manufacturers estimate for maintenance cost of this arrangement is C$ 75 000 per annum.

- Option C ('All-Steam' Plant)

The manufacturers estimates for maintenance cost of this plant is C$ 98 750 per annum.

The basic requirements of maintenance for the steam raising plant is briefly covered in Section 5.5.

6.3 Cost of Make-Up Fuel and Maintenance for a Ten Year Operational Period

The results of a costing comparison of make-up fuel and maintenance requirements for the COGAS and 'All-steam' plant for a 10 year operational period is as follows:

With LNG fuel make-up:

	Cost in C$, make-up fuel and maintenance
'All-Steam' Plant	31 110 660
COGAS	10 190 250
COGAS System Cost Saving	20 920 410

With diesel or fuel oil make-up:

	Cost in C$, make-up fuel and maintenance
'All-Steam' Plant	24 604 950
COGAS	13 404 720
COGAS System Cost Saving	11 200 230

Pro-rata cost saving would apply up to 20 years operation, subject to inflation. Assumed currency rates of exchange to mid June 1984 economics of:

1 USA $ = 1,3 Canadian $
1 pound sterling = 1,8 Canadian $

6.4 Manning

A gas turbine ship will require a smaller crew than a comparable steam turbine vessel. The 'all-steam' plant discussed in this study is estimated to require an engineering department complement of about 11 men out of a total crew of perhaps 30. The breakdown would be as follows:

Chief Engineer	1
1st Assistant Engineer	1
2nd Assistant Engineer	1
3rd Assistant Engineer	2
Electrician	1
Mechanic	1

Oiler	3
Wiper	1
TOTAL	11

The COGAS propulsion plant is designed for unattended operation at all times. Engine department personnel will work a normal eight-hour day performing routine tasks but no watches will be stood. The high level of automation of the gas turbine and the use of a 'slaved' steam turbine and non-supplementary fired recovery boiler make possible safe operation of the vessel by the absolute minimum number of crew.

A reasonable engineering complement for the gas turbine based propulsion system would be as follows:

Chief Engineer	1
1st Assistant Engineer	1
3rd Assistant Engineer	1
Electrician	1
Mechanic	1
Wiper	1
TOTAL	6

The reduction of the engineering complement by five men may mean a possible reduction in the steward department by one man bringing the total number of crew down to 24.

The reheat steam plant is more efficient than the more usual non-reheat plant but also more difficult to automate satisfactorily. The reheat type of steam plant, because of its many unattended pieces of equipment, has usually been denied operation with the minimum number of personnel. Two-man watches are stood by the 3rd Assistant Engineer and in this respect his responsibilities are greater compared with the same position aboard the COGAS vessel.

CANADIAN MANUFACTURING CONTENT

MANUFACTURER	TYPE OF PLANT	CANADIAN MANUFACTURING CONTENT
Rolls-Royce Canada Ltd.	RB 211/RT 56 Gas Turbine	50%
Foster Wheeler Power Products Ltd.	COGAS recovery boiler	85%
	All-steam reheat boiler	85%
Sunrod AB Sweden	COGAS recovery boiler	Prepared to source all parts other than boilers themselves in Canada
	All-steam reheat boiler	
Senior Green Ltd.	COGAS recovery boiler	Can confirm a high proportion of Canadian manufacture
N.E.I.	COGAS recovery boiler	Licensing arrangements under discussion with Canadian Manufacturers
G.E.C.Turbine Generators Ltd.	COGAS steam turbine	35%
	All-steam reheat turbine	35%
Westinghouse Canada	COGAS steam turbine	100%

Figure 37 CANADIAN MANUFACTURING CONTENT

FUEL CONSUMPTION AND COST,
LOADED LEG/BALLAST LEG
240 HOURS @ 17,5 KNOTS, 9 DEG C

	Small COGAS	Large COGAS	All-Steam Reheat
Total LNG ship requirement (all-purpose) T/h	3,764/3,59	3,674/3,497	5,086/4,78
Boil-off LNG flow T/h	3,375/2,7	3,375/2,7	3,375/2,7
LNG forced vaporisation for make-up T/h	0,389/0,890	0,299/0,797	1,711/2,080
Cost of make-up US $	18159/41545	13957/37204	79870/97094
Marine diesel make-up (COGAS) T/h or Fuel oil make-up 'All-Steam' T/h	0,446/1,022	0,343/0,915	1,877/2,282
Cost of make-up US $	24833/56905	19098/50947	62617/76128

Figure 38 FUEL CONSUMPTION AND COST FOR LOADED AND BALLAST LEGS

ANNUAL CONSUMPTION PER SHIP BASED ON 13 TRIPS PER YEAR METRIC TONNES

	Small COGAS	Large COGAS	All-Steam Reheat
Make-up from vaporised LNG	3991	3420	11 828
Make-up from marine diesel (COGAS) or Make-up from fuel oil 'All-Steam'	4580	3925	12 976

ANNUAL COST PER SHIP BASED ON 13 TRIPS PER YEAR US $

	Small COGAS	Large COGAS	All-Steam Plant
Make-up from vaporised LNG	776 152	665 093	2 300 532
Make-up from marine diesel (COGAS) or Make-up from fuel oil 'All-Steam'	1 062 594	910 585	1 803 685

BASED ON MDO @ 232 US $ Tonne MFO @ 139 US $ Tonne
LNG (FOB) @ 194,5 US $ Tonne

Figure 39 ANNUAL FUEL COSTS AND CONSUMPTION PER SHIP

7. CONCLUSIONS AND RECOMMENDATIONS

- CONCLUSIONS

Two sensible and perfectly feasible propulsion machinery solutions have been investigated.

The 'all-steam' plant which uses a reheat steam boiler and turbine represents current state of the art of this type of technology. The reheat system is well proven and is the modern solution for LNG carrier propulsion.

The RB.211 based gas turbine combined cycle plant is a slightly novel but nevertheless competitive alternative propulsion solution. The COGAS plant combines well proven gas turbine engineering with established waste heat recovery and steam turbine technology.

This study has considered the practicalities of the propulsion system beginning with the formulation of conceptual designs to provide a framework around which to develop engineering specifications, to obtain detailed engineering responses and financial guidance and finally to assess the cost competitiveness of the two main proposals.

The 'all-steam' plant uses standard equipment, although advanced designs using greater temperatures and pressures yielding better cycle efficiencies are available. These types of steam plant are not yet proven and so were not considered to be a practical solution to the problem.

The efficiency of the COGAS system is mostly dependant upon the extent to which the waste heat boiler is able to recover heat from the gas turbine exhaust. This ability is dictated by the tube area in the boiler which is in turn governed by the limits of a practical size of plant. By careful design of the boiler in order to maximise the tail cycle efficiency a good overall efficiency can be achieved with moderate plant size. This type of approach has been used in the study to provide a reliable and practical design solution for a plant which will meet the requirements of the ship.

Both types of plant offer good levels of redundancy in case of failure of items of the main propulsion plant. In the all-steam option shutdown of the main reheat boiler will mean that the auxiliary boiler will supply steam to the H.P. stages of the main steam turbine to get the ship home. The situation is slightly improved for the COGAS system since complete shutdown of either the steam turbine or the gas turbine can be tolerated. In this event the ship will still be capable of the 11 knots take-home condition.

As mentioned in the section on Plant Installation, discussion of the precise type of gearbox, it's cost and overhaul aspects, was not included as part of this study. Two preliminary designs were made as part of the work involved in the preparation of the

installation drawings for the 'all-steam' and COGAS plants, but more work will need to be carried out as part of a further study.

Fuel cost calculations clearly demonstrate the savings of the COGAS type of plant as compared with the reheat design. The actual savings depend upon the type of fuel used for supplementing the natural boil-off (only necessary during open water sailing) and the relative costs of these fuels. With LNG as the only type of fuel used onboard the annual saving per ship is estimated to be 1,6 Million US$ (price does not take account of fuel required to force boil the LNG for make-up). If it is not intended to force boil the cargo LNG then the dual fuel COGAS plant will use marine diesel as the make-up fuel.

Under the same conditions the reheat plant is capable of burning less expensive marine fuel oil as make-up instead of vapourised LNG and clearly this would be the most economical solution if an 'all-steam' plant were chosen.

Although the COGAS plant requires marine diesel as make-up (an expensive fuel), the annual saving is still estimated to be about 0,8M US$ compared with the reheat plant using fuel oil as make-up.

The large COGAS design using LNG as the make-up fuel therefore represents the most fuel economic means of ship propulsion investigated during this study.

- RECOMMENDATIONS

This study has been limited to investigating performance and cost aspects of a combined cycle gas turbine based plant as compared with a conventional reheat steam plant. Sufficient work has been done to show that the COGAS plant is feasible and offers both potential operating savings and the possibility of a significant amount of high technology work in Canada.

A COGAS plant could confidently be selected for any fugure LNG carrier of the type considered in the study. However there are a number of areas in which further investigation would be helpful in providing a basis for preparation of the specification for such a vessel. These include

1. Comparison of controllable pitch propeller systems with alternative reversing gearbox systems.

2. Comparison of alternative means of providing ship's electric power.

3. More detailed studies of maintenance, overhaul and manning costs for the complete machinery plant.

4. Analysis of the control requirements of a combined cycle plant

and simulation of the overall plant dynamic performance.

5. An overall ship study to check the interaction with the ship design of the different machinery systems to give an overall cost/benefit analysis.

Rolls-Royce (Canada) Ltd would be pleased to discuss participation in further studies to investigate these and other relevant areas.

REFERENCES

[1] Spears H.C.K., "Advanced Systems for the Gas Turbine, Steam, and Combined Cycle Propulsion of Ships". Centenary Conference on Marine Propulsion, Paper No. 11, May 1984.

[2] Marron H.D., "U.S. Navy Racer Development Programme". ASME Paper 84 - GT-100.

BIBLIOGRAPHY

Bandat N.P. & James O.R., "Application of a Power Recovery System to Gas Turbine Exhaust Gases", ASME Paper 79-GT-167.

de Biasi V., "SK30 Total Energy Plant Rated at 73% Efficiency", Gas Turbine World, July 1980.

Burnside B.M., "Improved Dual Pressure Bottoming Cycle for Waste Heat Recovery", Trans. I Mar E, Vol.93, 1981.

Coats R., "Modern Marine Steam and Gas Turbines for Merchant Ships", Marine Engineers Review, December 1971.

Fiminia J., "Economic Comparison of Various Marine Power Plants". Presented to SNAME Annual Meeting, New York, November 1973.

Gerstin R.A., "A Technical and Economic Overview of the Benefits of Repowering", ASME Paper 75-GT-16.

Gibbon R.P. & Rolih I.H., "COGAS - Marine Power Plant for Energy Savings", Technology Vol. 16, No.3, July 1979.

Halkolas J.T., Campbell A.H. & Jung D., "Racer Conceptual Design", ASME Paper 83-GT-50.

Hauschildts M.R., Gripp L.P. & Peters W., "An LNG Carrier Adjustable Speed Turbine Electric Propulsion System", ASME Paper 81-GT-112.

Howard J.L. & Kuamsdal R.S., "Energy Efficient LNG Carriers", SNAME Symposium on Ship Costs and Energy, October 1982.

Katz Y. & Boyen J.L., "Design Considerations for Heat Recovery System for DD-963 Class Ship", ASME Paper 77-GT-106.

MacIntyre N.L., "Design Aspects of an Advanced 14 000 H.P Marine Combined Cycle Plant

Mangan J.L. & Pettit, R.C., "A Highly Efficient Steam Turbine - Gas Turbine Cycle", ASME Paper 63-AHGT-51.

Marron H.D., "Gas Turbine Waste Heat Recovery Propulsion for U.S. Navy Surface Combatants", Naval Engineers Journal, October 1981.

Marron H.D. & Carleton R.S., "The Gas Turbine Waste Heat Recovery System and the U.S. Navy", ASME Paper 78-GT-170.

Mattson W.E., "Shipboard Energy Savings with the Racer System", San Diego Section of the American Society of Naval Engineers, May 1980.

Marwood R.M. & Bassilakis C.A., "The Thermodynamic Design of a Combined Steam and Gas Turbine Marine Propulsion System", ASME Paper 67-GT-16.

Merz C.A. & Pakula T.J., "The Design and Operational Characteristics of a Combined Cycle Marine Power Plant", ASME Paper 72-GT-90.

Miwa K., "Trend of Gas Turbines and Combined Cycle Plants for Marine Use", Journal of MESJ Vol. 6, No. 2 Bulletin, June 1978.

Morgan D.T. & Davies J.P., "High Efficiency Gas Turbine/Organic Rankine Cycle Combined Power Plant", ASME Paper 74-GT-35.

Morton Prof. A.J., "Thermodynamics of Waste Heat Recovery in Motor Ships", Trans. I.Mar.E. 1981 Vol 93 (Paper C69).

Ohashi S. & Suzuki T., "Recovery of Waste Heat by Organic Fluid", Journal of MESJ, Vol. 6, No. 2 - Bulletin, June 1978

Ondryas, I.S., "Fuel Profitability and Design of Gas Turbine Cogeneration Plant", ASME Paper 82-GT-142.

Pringle, G.G., "Economic Power Generation at Sea: The Constant Speed Shaft Generator", Trans. I.Mar.E., May 1982.

Sharp Inc., George G., "Applicability of COGAS Marine Power Plants for Merchant Ships", Report No. MA-RD-920-7901 for Contract No. 7-3800 on behalf of U.S. Department of Commerce - Maritime Administration, January 1979.

Sheldon R.C. & Todd D.M., "Optimization of the Gas Turbine Exhaust Heat Recovery System", ASME Paper 71-GT-79

Stambler I., "New Breed of Combined Cycle Plants 4 to 11 MW's", Gas Turbine World, July 1980.

Stott C.W. & Casey J.P., "An Updated Analysis if Fuel Consumption: Steam and Diesel", Marine Technology Vol. 17, No.4, October 1980.

Svenson N.A., "A Gas Turbine Powered LNG Tanker", ASME Paper 73-GT-27.

Tintori J., Schiefer R.B., & Taylor J.R., "A Fuel for Total Energy", ASME Paper 71-GT-55.

Witt K., "Onboard Reliquefaction of LNG Boil-Off", Trans. I.Mar.E., October 1980.